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PRACTICAL GUIDE FOR SHIPBOARD VIBRATION CONTROL AND ATTENUATION

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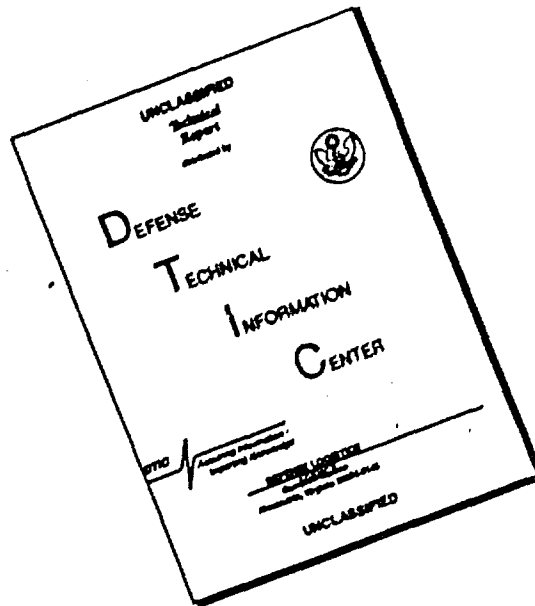
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Although ship designers make every effort to prevent objectionable and detrimental vibrations before a vessel is constructed, there are times when vibrations are at an unacceptable level after construction or major modification.

This volume is intended to aid shipboard personnel in locating the source of unacceptable vibrations and to guide them in curing their own problem. A chapter giving typical examples is included for this purpose. With this guide in hand, shipboard personnel will be better able to know when they need to call on expert assistance should the need arise. For shipboard personnel who wish to know more about vibration theory, an appendix is included.

CLYDE L. LUSK, JR.
Rear Admiral, U.S. Coast Guard
Chairman, Ship Structure Committee

| | | | | | |
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METRIC CONVERSION FACTORS

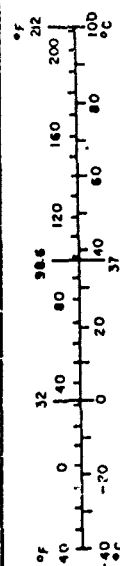
Approximate Conversions to Metric Measures

| Symbol | When You Know | Multiply by | To Find | Symbol |
|----------------------------|------------------------|----------------------------|---------------------|-----------------|
| LENGTH | | | | |
| in | inches | 2.5 | centimeters | cm |
| ft | feet | 30 | centimeters | cm |
| yd | yards | 0.9 | meters | m |
| mi | miles | 1.6 | kilometers | km |
| AREA | | | | |
| in ² | square inches | 6.5 | square centimeters | cm ² |
| ft ² | square feet | 0.09 | square meters | m ² |
| yd ² | square yards | 0.8 | square meters | m ² |
| mi ² | square miles | 2.6 | square kilometers | km ² |
| | acres | 0.4 | hectares | ha |
| MASS (weight) | | | | |
| oz | ounces | 28 | grams | g |
| lb | pounds | 0.45 | kilograms | kg |
| | short tons (2000 lb) | 0.9 | tonnes | t |
| VOLUME | | | | |
| tsp | teaspoons | 5 | milliliters | ml |
| Tbsp | tablespoons | 15 | milliliters | ml |
| fl oz | fluid ounces | 30 | milliliters | ml |
| c | cups | 0.24 | liters | l |
| pt | pints | 0.47 | liters | l |
| qt | quarts | 0.95 | liters | l |
| gal | gallons | 3.8 | liters | l |
| fl ³ | cubic feet | 0.03 | cubic meters | m ³ |
| yd ³ | cubic yards | 0.76 | cubic meters | m ³ |
| TEMPERATURE (exact) | | | | |
| °F | Fahrenheit temperature | 5/9 (after subtracting 32) | Celsius temperature | °C |

* 1 m = 2.54 in exactly. For other exact conversions and more detailed tables, see NBS NISTC Publ. 286, Units of Weights and Measures, Price \$2.25, SO (along No. C13) TO 286.

Approximate Conversions from Metric Measures

| Symbol | When You Know | Multiply by | To Find | Symbol |
|----------------------------|-----------------------------------|-------------------|------------------------|-----------------|
| LENGTH | | | | |
| mm | millimeters | 0.04 | inches | in |
| cm | centimeters | 0.4 | inches | in |
| m | meters | 3.3 | feet | ft |
| m | meters | 1.1 | yards | yd |
| km | kilometers | 0.6 | miles | mi |
| AREA | | | | |
| cm ² | square centimeters | 0.16 | square inches | in ² |
| m ² | square meters | 1.2 | square yards | yd ² |
| km ² | square kilometers | 0.4 | square miles | mi ² |
| ha | hectares (10,000 m ²) | 2.5 | acres | |
| MASS (weight) | | | | |
| g | grams | 0.035 | ounces | oz |
| kg | kilograms | 2.2 | pounds | lb |
| t | tonnes (1000 kg) | 1.1 | short tons | |
| VOLUME | | | | |
| ml | milliliters | 0.03 | fluid ounces | fl oz |
| l | liters | 2.1 | pints | pt |
| l | liters | 1.06 | quarts | qt |
| l | liters | 0.26 | gallons | gal |
| m ³ | cubic meters | 36 | cubic feet | ft ³ |
| m ³ | cubic meters | 1.3 | cubic yards | yd ³ |
| TEMPERATURE (exact) | | | | |
| °C | Celsius temperature | 9/5 (then add 32) | Fahrenheit temperature | °F |



PREFACE

The purpose of this Guide is to provide operators, shipyards, ship owners, and others who must deal with ship vibration problems, but who have limited knowledge and experience in the field, with an understanding of the nature of the most common problems frequently encountered, how to assess and evaluate them, and what alternatives are available for corrective action. Basic information on vibration fundamentals, instrumentation, measurement procedures, and vibration criteria is included. Typical examples of shipboard vibration problems encountered and their solutions are also presented.

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SECTION 1.0

INTRODUCTION

Vibration aboard ship can result in fatigue failure of structural members or major machinery components, can adversely affect the performance of vital shipboard equipment and increase maintenance costs, and result in discomfort or annoyance to passengers and crew. Generally, hull vibration will be identified as objectionable to the crew before it becomes damaging to the ship's structure. Failure of major machinery components and vital shipboard equipment, however, can occur without significant annoyance to those aboard the ship.

A ship is an extremely complex assembly of structural and mechanical components which are, in turn, stimulated by a large number of dynamic forces both transient and periodic in nature which may be significantly increased in severity by sea and operating conditions. The response of the total structural and mechanical systems to the broad range of exciting forces and poorly defined damping characteristics would seem to result in problems of infinite dimensions to the designer. As a result, although much research has been carried out since World War II, it has generally been fragmentary in nature or related to attempts to solve individual problems as they arise. Although limited vibration studies are normally conducted during the design and construction of most ships, the complexity of the many potential problems can result in serious shipboard vibration problems. The occurrence of such problems, once a ship has put to sea, therefore, is not unusual. In addition, service problems may develop during normal ship operations.

The purpose of this Guide is to provide operators, shipyards, ship owners, and others who must deal with ship vibration problems, but who have limited knowledge and experience in the field, with an understanding of the nature of the most common problems frequently encountered, how to assess and evaluate them, and what alternatives are available for corrective action. It must be recognized, however, that suitable vibration measuring instrumentation is normally required, and tests must be conducted under controlled conditions to obtain useful data for purposes of evaluation and interpretation against acceptable design criteria.

Experienced vibration test engineers are required to conduct the necessary investigations in all but the simplest problems encountered. It should also be pointed out that there is a large gap between the identification of the problem and the development of the optimum corrective action because of the analyses and/or laboratory studies required and the constraints existing in a completed ship. In many cases, however, the guide can be effectively used to solve many of the more common local vibration problems and to provide adequate information to identify the problem and to direct the technical effort required for the more complex problems.

The procedures and recommendations given are based on experience and, whenever possible, relate to existing criteria, standards, and test codes in current use. Adequate references are provided for further in-depth study.

Typical examples of frequently encountered problems and their solutions are included. For convenience, commonly used vibration terms, definitions, a limited review of vibration fundamentals, and suitable background data on shipboard vibration is provided.

SECTION 2.0

SCOPE AND FIELD OF APPLICATION

This publication provides guidance for the measurement, evaluation, and control of shipboard vibration in single- or multiple-shaft ships. The term shipboard vibration, as used in this guide, relates to the periodic forces generated by wave excitation by the ship's propellers, the main propulsion machinery, or other shipboard machinery, and the response of the ship's structure, structural components, machinery, or shipboard equipment to these exciting forces. It does not relate to transient excitation resulting from wave impact or slamming of the ship.

This publication relates to the control and attenuation of those design factors (exciting forces or dynamic response characteristics) which are built into the ship and which contribute to alternating stresses or a vibratory environment which may prove to be unsatisfactory to the ship's structure, machinery, equipment, or personnel. It will also apply to an increase in vibration resulting from damage, such as bent propeller, or maintenance problems, such as excessive bearing wear down.

This publication is primarily concerned with:

- Hull vibration
- Vibration of propulsion systems as it affects hull vibration or the integrity of the propulsion system itself
- Vibration of major structural assemblies
- Vibration of local structures and shipboard equipment
- Vibration as it affects shipboard personnel.

Whenever possible, generally accepted criteria or standards, relative to the measurement and evaluation of shipboard vibration, will be referenced.

Although the procedures, measurement techniques, and criteria employed have been developed for seagoing ships 200 feet or greater in length, they can be employed, in most instances, for inland ships and tugboats.

SECTION 3.0

INTRODUCTION TO SHIP VIBRATION - EXCITATION AND RESPONSE

A ship may be subjected to rigid body motions (i.e., roll, pitch, heave, etc.), to transient dynamic forces imposed by random seas, maneuvering, or slamming due to bow motions in sea waves, and to periodic excitation due to hydrodynamics of flow over appendages, the pressure fluctuations due to propellers, and machinery forces due to unbalance or engine harmonics generated by diesel engines. Although both rigid body motions and transient dynamic forces will influence ship vibration, this guide will specifically address the response of the structural and mechanical components of the ship as influenced by the periodic forces present. The significance of the periodic forces present and the dynamic response of the ship or its components are generally built in during the design phase (intentionally or unintentionally). Design studies do not normally include the imposition of transient excitation forces in vibration analyses, and tests are conducted in such a manner as to minimize these transient forces.

In the development of this guide an effort is made to present sufficient information to understand the basis for the generally observed vibration phenomenon. It is recommended that for a more in-depth understanding, however, that the reader make use of the selected references given. It is suggested that "Ship Hull Vibration" [3-1], "Ship Vibration" [3-2], and "Mechanical Vibrations" [3-3] be referred to for a more complete understanding of ship vibration.

3.1 BACKGROUND

While it is theoretically true that all mechanical systems present in the total ship and all dynamic forces do contribute to the total ship response, the study of the complete system as a single unit will normally be found to be impractical and unnecessary. Typical examples include torsional vibration of the propulsion system which rarely influences hull vibration; dynamic unbalance of small machines may adversely effect the local structures to which it is attached but is not likely to influence hull vibration while unbalanced forces of large prime movers may; vibration of shipboard equipment is principally controlled by the way it is designed and installed and, although it may be excited by the same forces that vibrate the hull, the vibration of the equipment generally will not affect the hull response.

In an attempt to simplify what could be a most complex problem, it becomes prudent to reduce it to more practical considerations. As an approach, therefore, it has been found to be convenient in both design studies and shipboard evaluation to divide the complete ship system into the following basic elements:

- Hull Girder
- Major Structural Substructures
- Local Structural Elements
- Shipboard Equipment
- Main Propulsion Machinery Systems

Considering the ship in this light is particularly helpful in the diagnosis, evaluation, and development of corrective action in the resolution of shipboard vibration problems.

The first three elements are structural, are in descending order of size, are primarily excited by propeller forces transmitted through the structures, and respond directly to the applied forces as transmitted by the intervening structure.

Shipboard equipment is classified as active when it generates vibratory forces or passive if it does not. A generator set is active and an electrical transformer is passive. The response of shipboard equipment may be related to its own exciting forces or to those transmitted through the ship's structure.

The main propulsion machinery system may be excited by the ship's propeller, by dynamic or hydrodynamic unbalance, or, in the case of diesel engine applications, by harmonics of the engine. The vibration of the machinery system can prove to be damaging to the hull structure, equipment, or to the machinery system itself.

An understanding of the excitation and response of these individual elements and their interrelationship will assist in the diagnosis of most vibration problems encountered. Each of the five elements are treated in greater depth in the following sections.

3.2 HULL GIRDER VIBRATION

The ship's hull girder includes the shell plating, main deck, and all internal members which collectively provide the necessary strength to satisfactorily perform the design functions of the ship in the expected sea environment. The hull girder responds as a free-free beam (both ends free) when subjected to dynamic loads. Although the surrounding water and loading of the hull influences its response, the hull girder will always respond as a free-free beam. Vibration of the hull girder, excited by alternating propeller forces, represents the most frequent source of troublesome vibration encountered aboard ship. The vibration characteristics of the ship are primarily established by the propeller and stern configuration. After the ship is built, modifications to correct excessive vibration resulting from improper propeller and/or stern lines are generally most extensive, expensive, and impractical. In addition,

vibration of the hull girder will provide the excitation to the major substructures, local structural elements, and shipboard equipment. Main propulsion machinery and auxiliary machinery can also contribute to general hull vibration and the vibration of local structural components.

A ship's hull girder responds in vertical flexure when subjected to wave impact. In oceangoing ships subjected to random seas, the dynamic response at the fundamental natural frequency of the hull is normally at low stress levels and is referred to as transient in nature and, as indicated in Section 2.0, is not treated in this publication. In the case of ore carriers on the Great Lakes, however, periodic vibration of the hull girder at its fundamental natural frequency has been found to be a potentially dangerous structural problem and is referred to as Springing. Further discussion on Springing is given in Sections 5.0 and 7.0.

3.2.1 Hull Girder Excitation

Dynamic forces entering the hull through the propulsion shaft bearings or directly through pressure forces impinging against the hull, generated by the propeller blades, are by far the main source of hull girder vibration. In the case of slow-speed diesel engine drive systems, engine unbalanced or firing forces may also be important. Less important sources are auxiliary machinery and hydrodynamically excited appendage vibration. When attempting to determine the source of vibration it is necessary to determine the frequency of excitation and it is convenient to relate it to the shaft rotational frequency by determining the number of oscillations per shaft revolution (order). The total signature may include first order, blade-frequency, harmonics of blade frequency, as well as constant frequency components.

3.2.1.1 Shaft Frequency Forces

Mechanical forces which are associated with shaft rotational speed (1st order) may result from one or more of the following causes:

- a. Shaft unbalance
- b. Propeller unbalance
- c. Propeller pitch error
- d. Engine unbalance (for slow-speed diesel driven ships)
- e. Bent shafting
- f. Journal eccentricity
- g. Coupling or flange misalignment

The most likely causes of shaft frequency forces are those due to a, b, c, and d above. The other possible causes are not as likely to occur if

reasonable specifications, workmanship, and inspection procedures are exercised during the design and construction of the ship.

Shaft frequency forces occur within a low frequency range. They are, however, of considerable concern since they may be of large magnitude and may excite one of the lower hull modes at or near full power, thus producing a significant resonance effect.

The principal engine unbalance encountered with slow-speed diesel driven ships are the primary and secondary free engine forces and moments. Of particular concern is the magnitude of the forces and moments, the location of the engine, and the possible correlation of these inputs with the lower vertical and athwartship natural frequencies of the hull girder. The primary forces and moments occur at shaft frequency and the secondary forces and moments occur at twice shaft frequency. The magnitude of these forces and moments should be furnished by the engine builder. For additional information on diesel engine forces as they effect hull vibration, see Section 6.5.5.

3.2.1.2 Propeller Forces

In addition to the basic design purpose of generating steady thrust for the ship's propulsion, the marine propeller also generates undesired fluctuating dynamic forces and moments due to its operation in a nonuniform wake and due to the passage of the blades close to the hull and appendages. These fluctuating forces and moments are usually referred to as propeller forces and are of blade frequency and its higher harmonics. The higher harmonics are normally of secondary importance. These propeller forces are, in turn, categorized into two groups, the bearing forces and the hull pressure forces.

A more detailed description of the alternating forces generated by a ship's propeller may be obtained in "Principles of Naval Architecture," published by S.N.A.M.E. and the many papers presented on the subject in recent years. However, for purposes of this guide, it would be helpful to provide some physical insight on how a propeller generates the unsteady forces and moments.

Propeller theory relates to its operation "in open water," in which it is advancing into undisturbed water. However, when it is operating behind the hull it is working in water which has been disturbed by the passage of the hull and the water around the stem has acquired a forward motion in the same direction as the ship. This forward moving water is called the wake and it varies in the plane of the propeller disc, giving rise to variations in loading on each blade as the propeller rotates.

Since the propeller produces both torque and thrust, both components vary with each blade as it passes through the uneven wake, giving rise to alternating torque and thrust at propeller blade frequency and harmonics of blade frequency. As a further effect of the uneven loading of the propeller, the center of thrust is eccentric to the physical center of the propeller and creates bending moments in the shaft and vertical and transverse forces in the bearing [3-7]. These forces are also at blade frequency and harmonics of blade frequency while the principal bending stress in the shaft occurs at the shaft

frequency with smaller components at $n - 1$ and $n + 1$ order, where n = the number of propeller blades.

Similarly, alternating pressure forces are generated by the operation of the propeller blades adjacent to the hull surfaces in the axial and transverse directions. The total effect results in the forces and moments generated on the hull surface with reactions on the propeller blades producing bearing forces. To minimize these forces, maximum clearances are required in the axial (forward) directions and in the propeller tip clearance. The propeller generated hull pressure forces are greatly increased if cavitation exists [3-4]. The collapse of air pockets produce implussions, which are characterized by the hammering frequently noted in the stem compartments and the presence of higher harmonics of blade frequency.

3.2.1.2.1 Bearing Forces. The bearing forces originate from the nonuniformity of the wake in the plane of the propeller disc. The strength of the various harmonics of the wake affect the magnitude of the bearing forces and influence the choice of the number of propeller blades. The relative strength of the various orders of wake harmonics indicate the relative strength of the blade-frequency forces. The wake, in turn, is influenced by the design of the hull form. An optimum design of the hull form would reduce the nonuniformity of the wake, thereby reducing the magnitude of the bearing forces. The bearing forces excite the ship through the propulsion shafting/bearing system, and are fully described by six components as illustrated in Figure 3-1. As shown in Figure 3-1, with the origin of axes at the center of the propeller these components are the thrust and torque in and about the longitudinal or fore-aft axis; the horizontal bearing force and the vertical bending moment in and about the horizontal or athwartship axis; the vertical bearing force and horizontal bending moment in and about the vertical axis.

The vertical and horizontal bearing forces result from differences in torsional forces on the blades of the propeller, while the vertical and horizontal bending moments are due to the propeller thrust which occurs at a point which is eccentric to the center of the propeller.

3.2.1.2.2 Hull Pressure Forces. The hull pressure forces originate from the pressure variation caused by the passage of propeller blade tips close to the hull and appendages. The hull pressure forces are affected by propeller-hull clearance, by blade loading, and by changes in the local pressure field around the blade. The occurrence of blade cavitation will drastically increase the pressure forces. In some cases, a 20 to 40 times increase of hull pressure forces due to cavitation has been observed in experimental measurement, as compared to noncavitating condition [3-4]. The pressure forces excite the ship through the hull bottom surface in way of and adjacent to the propeller. The pressure forces are fully described by six components, namely the longitudinal force and moment in and about the fore-aft axis; the horizontal force and vertical moment in and about the athwartship axis; and the vertical force and horizontal moment in and about the vertical axis, as illustrated in Figure 3-2.

3.2.1.2.3 Effect of Propeller Forces. The alternating blade frequency thrust of the bearing forces provides the principal excitation to the propulsion

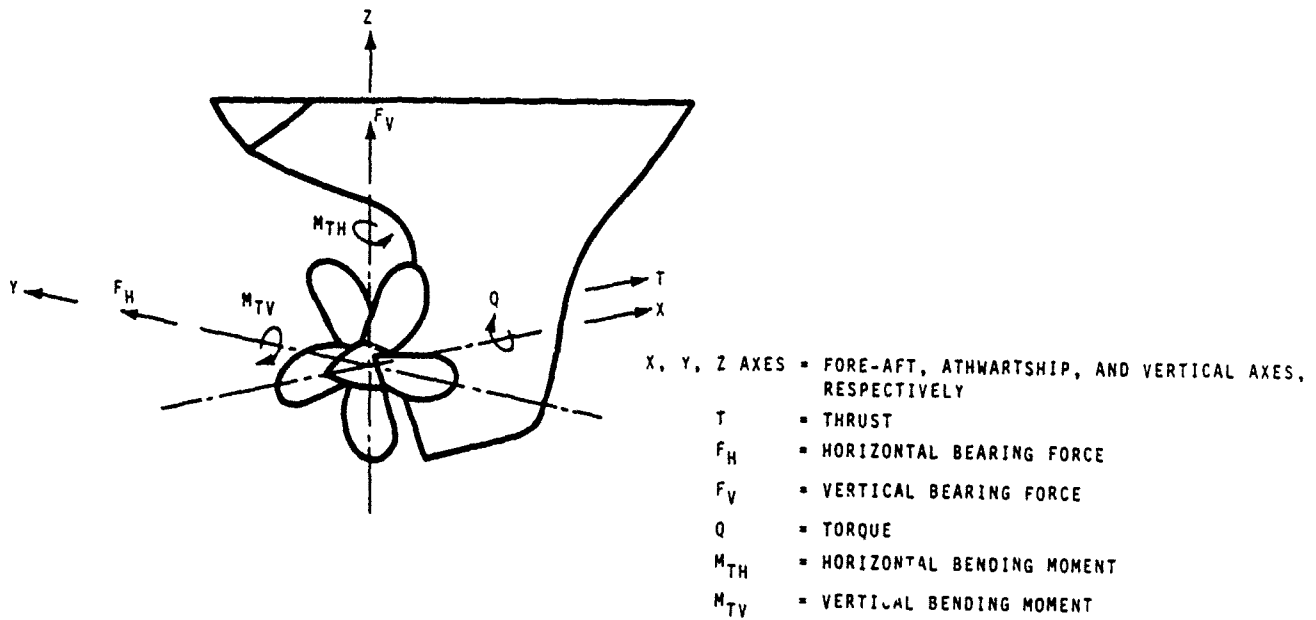


Figure 3-1. Description of Bearing Forces and Moments

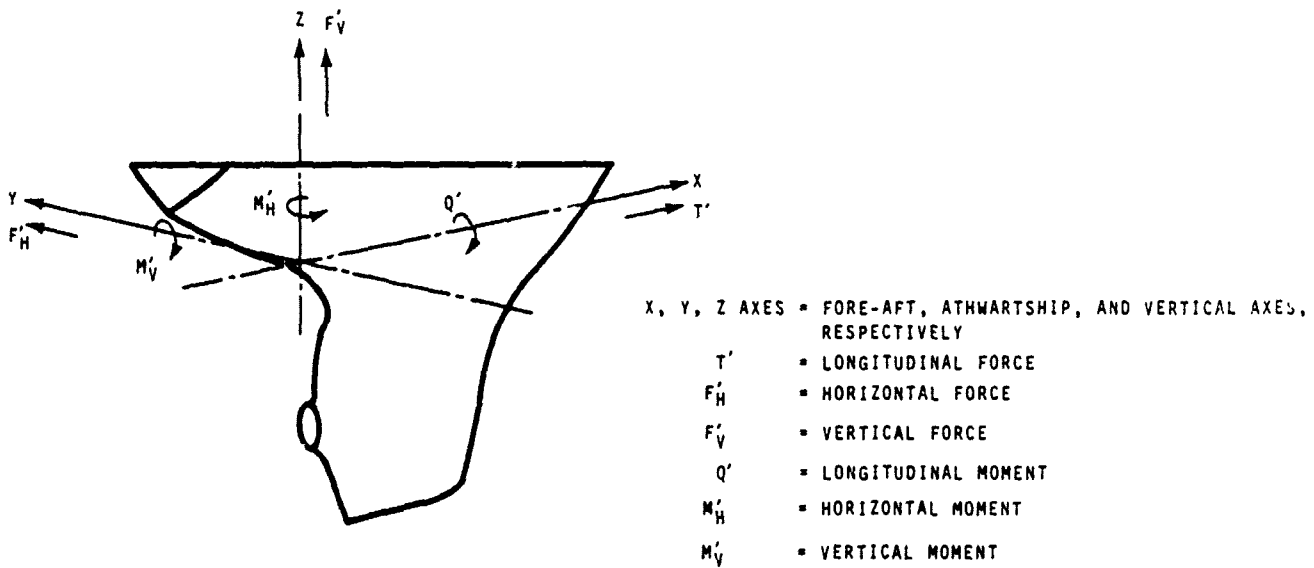


Figure 3-2. Description of Hull Pressure Forces and Moments

system in the longitudinal mode, while the blade frequency torque constitutes the principal excitation to the propulsion system in the torsional mode. The blade frequency vertical bearing force, when vectorily combined with the blade frequency vertical pressure force, provides the total vertical force which excites the hull in the vertical direction. Similarly, the horizontal bearing forces, when combined with the blade frequency horizontal pressure forces, provides the major contribution for exciting the hull in the horizontal direction. The vertical and horizontal forces and their distance from the neutral axis of the hull, combine to excite the hull torsionally. Longitudinal hull pressure forces and alternating thrust entering the hull through the thrust bearing will combine to excite the hull in the longitudinal direction.

3.2.2 Hull Girder Response

The response of the hull girder may be resonant or nonresonant (forced). It is likely to be resonant through the first five or six modes of vibration when driven by the shaft or propeller frequencies present. Above the fifth or sixth mode the hull girder vibrates approximately in proportion to the forces generated (forced vibration). Principal exciting frequencies are shaft frequency, propeller blade frequency, and harmonics of propeller blade frequency. Hydrodynamic forces may also stimulate the resonant frequency of the hull, rudders, or struts excited by hydrodynamic flow over the appendage.

3.2.2.1 Modes of Vibration of the Hull Girder

The hull girder will normally vibrate in the following modes:

- Vertical Flexure (Figure 3-3)
- Horizontal Flexure (Figure 3-4)
- Torsional (Twist) (Figure 3-5)
- Longitudinal (Compression) (Figure 3-6)

Coupling may exist between vertical and longitudinal and between horizontal and torsional modes. The most significant vibration is normally associated with vertical and horizontal flexure.

3.2.2.2 Frequency of Vibration of the Hull Girder

Vertical flexural hull vibrations is the most important type of resonant hull vibration which may be encountered in service. As previously noted, this may be excited by dynamic or hydrodynamic unbalance of the propeller, dynamic unbalance or eccentricity of shafting or other large rotating masses such as bull gears, and by primary or secondary unbalanced moments of direct drive diesel engines. Transient forces, introduced by sea waves, may also excite hull natural frequencies.

In twin screw ships significant excitation of horizontal modes may occur due to phasing of propeller unbalance forces.

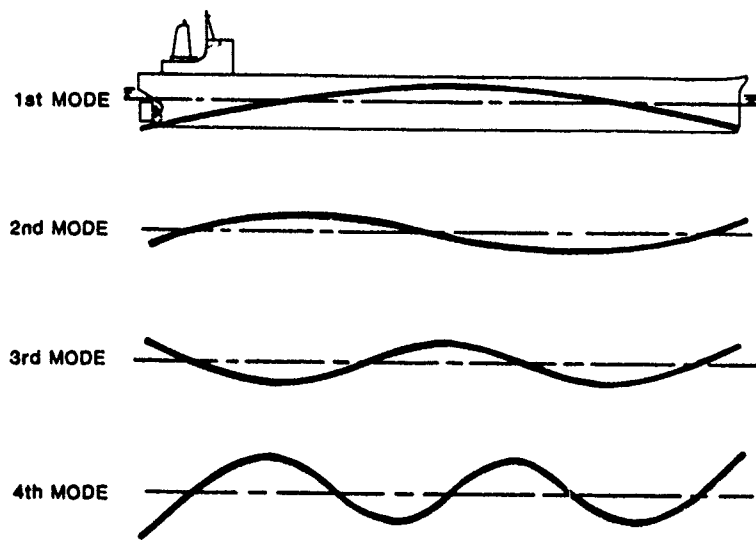


Figure 3-3. Hull Girder Vertical Vibration of 2-5 Nodes (1st - 4th Mode)

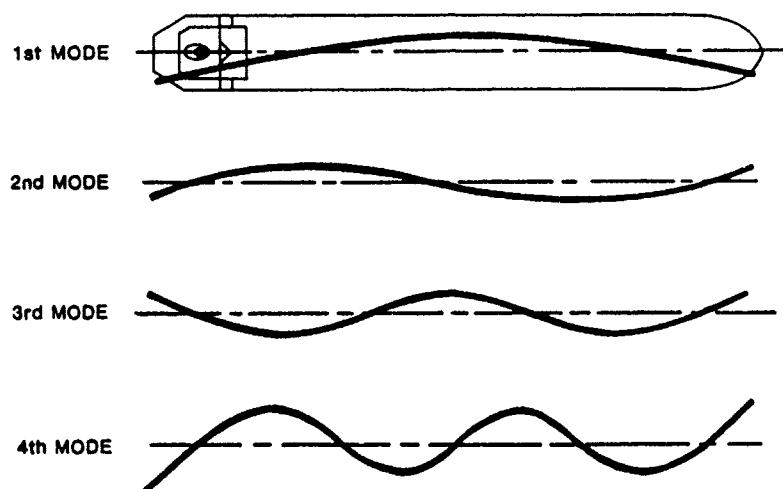


Figure 3-4. Hull Girder Horizontal Vibration of 2-5 Nodes (1st - 4th Mode)

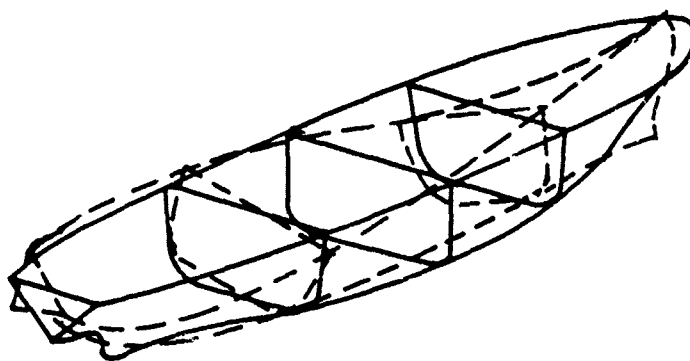


Figure 3-5. Hull Girder Torsional Vibration

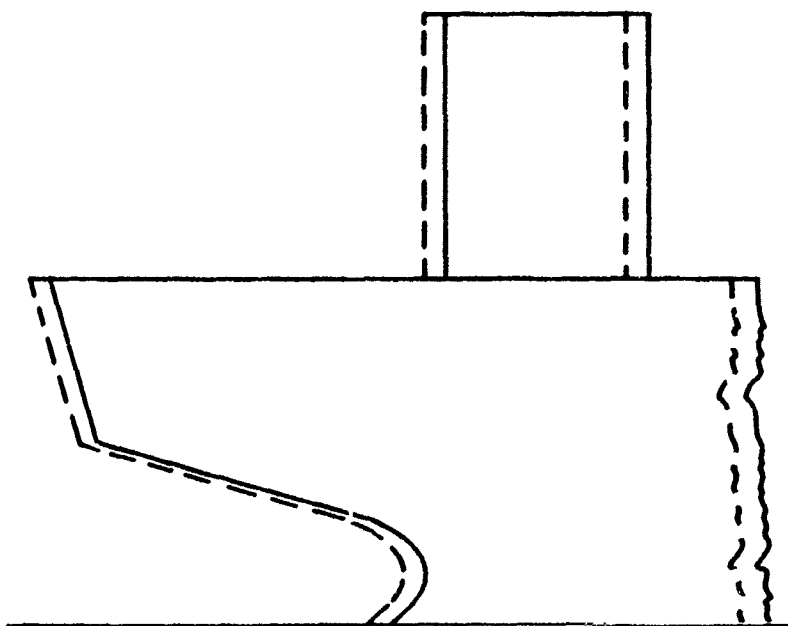


Figure 3-6. Hull Girder Longitudinal Vibration

Some ships, particularly container ships with large deck openings, may be sensitive to torsional response excited by horizontal propeller forces.

In diagnosing hull vibration problems, it is helpful to be able to estimate hull natural frequencies. A number of empirical relations have been developed for estimating the fundamental vertical natural frequency of ship hulls, N_{2v} . That developed by Todd [3-1] is:

$$N_{2v} = C_1 \sqrt{\frac{BD^3}{\Delta_1 L^3}} + C_2, \text{ cycles per minute}$$

where: B = Beam of ship, in feet

D = Depth of ship to strength deck, in feet

T = Draft, in feet

L = Length of ship between perpendiculars, in feet

Δ = Displacement, in long tons

$$\Delta_1 = \left(1.2 + \frac{B}{3T}\right) \Delta$$

Suggested constants are:

| | C_1 | C_2 |
|---------------|--------|-------|
| Tankers | 52,000 | 28 |
| Cargo Vessels | 46,750 | 25 |

for higher modes:

$$N_{nv} = N_{2v} (n - 1) \mu_v \text{ Hz}$$

where: $\mu_v = 1.02$ for tankers

1.0 for bulk carriers

0.845 for cargo ships

As a general rule, the higher modes will follow the fundamental frequency by the ratios 1, 2, 3, 4, etc. Vertical frequencies may be estimated from Figure 3-7 from Det Norske Veritas Guidelines [3-5]. Estimation of

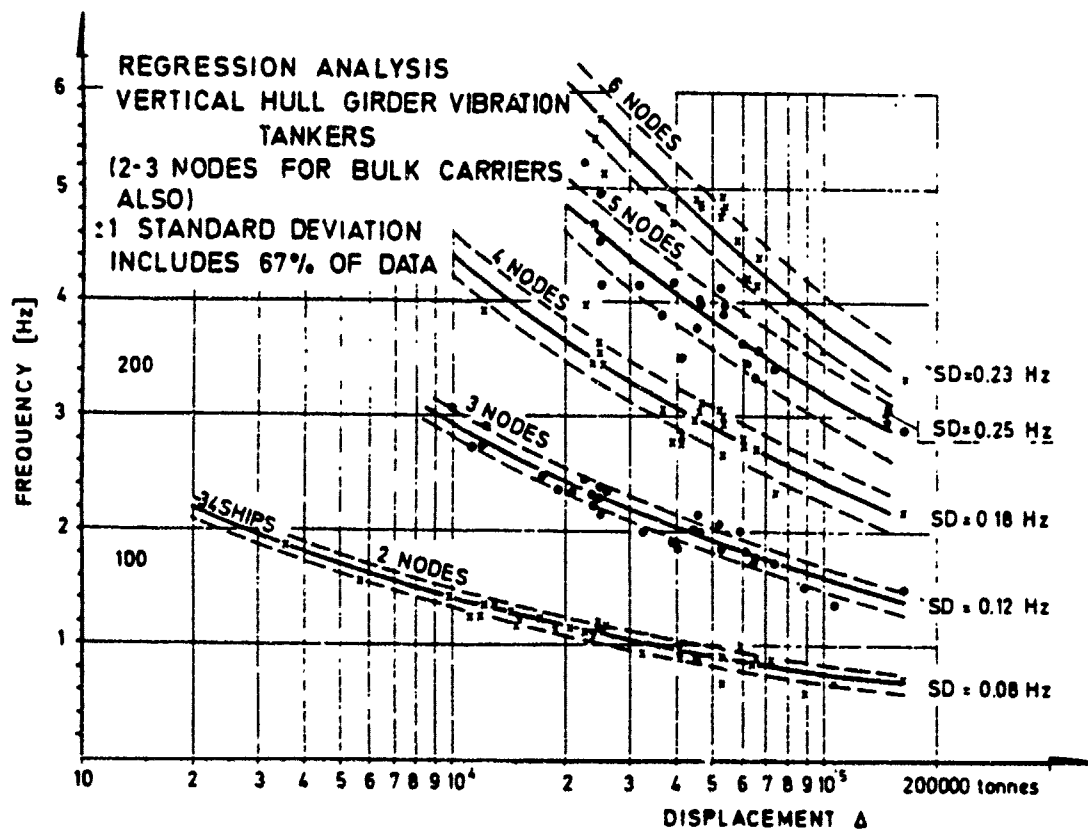
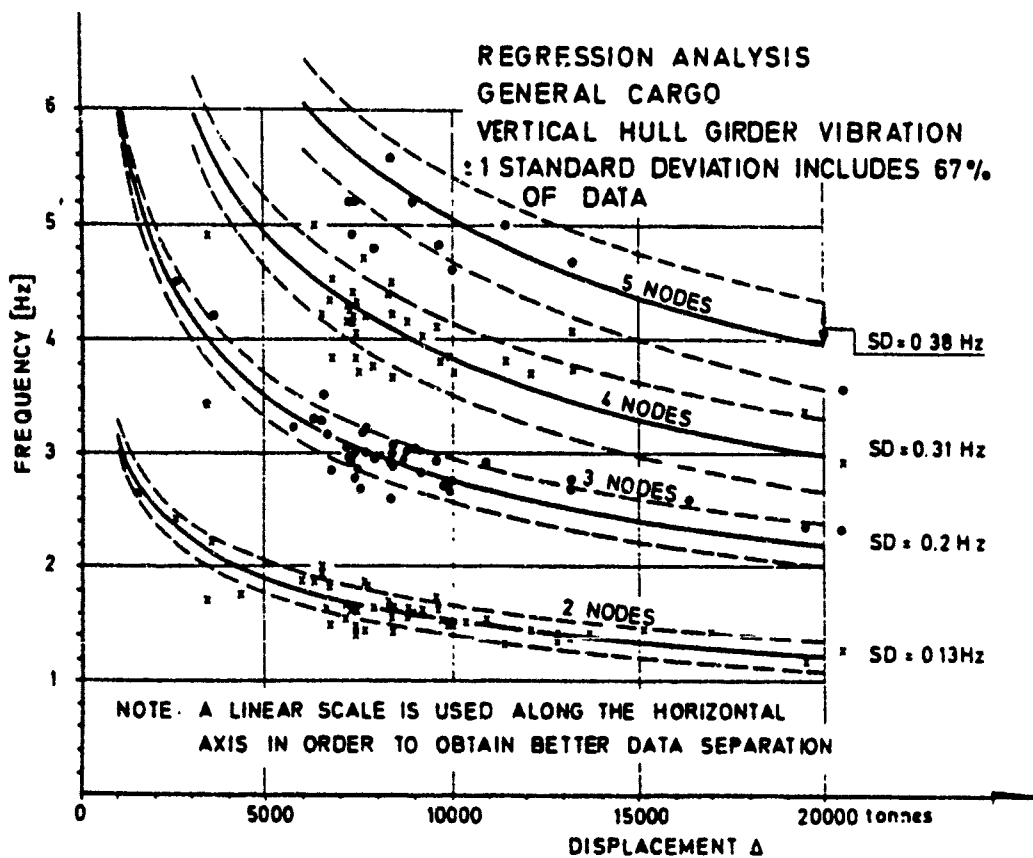


Figure 3-7. Plots of Hull Girder Vertical Vibration

vertical hull frequencies for the first five modes is considered reasonable for evaluation purposes during trial investigations. A check of the fundamental frequency may be made by dropping and snubbing the anchor.

Horizontal flexural frequencies follow a similar pattern. However, the fundamental (two noded) frequency will be approximately 50 percent higher than the fundamental vertical frequency.

The fundamental torsional mode of the hull girder may be estimated at approximately twice the horizontal or three times the first vertical natural frequency.

The longitudinal natural frequency may be estimated to be approximately three and one-half times the fundamental horizontal mode.

3.2.2.3 Effects of Adverse Operating Conditions

Adverse operating conditions frequently result in significant increase in vibration amplitudes. When reporting shipboard vibration, or responding to reported problems, it is extremely important to recognize that shipboard vibration is a somewhat random phenomenon and the operating conditions must be reported for the data given. This factor also has a significant impact on the analysis and reporting of the data which is used for evaluation purposes. Details are given under Section 6.0, Measurement Methods. Some relevant factors are given below.

3.2.2.3.1 Sea Conditions. Under ideal sea conditions (flat calm, straight ahead) hull vibration signals will modulate from maximum to minimum by a factor of 2 to 1.

Under prescribed trial conditions (sea state 3 or less) hull vibration signals may modulate by a factor of 3 to 1. Higher factors may exist under adverse weather conditions.

3.2.2.3.2 Hard Maneuvers. During hard turns amplitudes may readily increase by a factor of two for single screw ships and by a factor of three for twin screw ships.

During a crash-back (full ahead to full astern) the alternating thrust may exceed the driving thrust and could result in damage to the thrust bearings if care is not exercised. Before requesting this maneuver, it is prudent to check the procedure at lower speed conditions and to monitor the thrust bearing response throughout the operation.

3.2.2.3.3 Shallow Water. An increase in hull vibration by 50 percent may be experienced in shallow water. Shallow water in this context is a depth of less than six times the draft of the ship.

¹ This precautionary note is recommended for all sea trials.

3.2.2.3.4 Light Draft Condition. An increase in hull vibration by 25 percent may be experienced in ballast condition. For minimum hull vibration, full load with aft peak tanks filled is recommended.

3.3 VIBRATION OF MAJOR SUBSTRUCTURES

For purposes of evaluation, major substructures are defined as secondary structures of sufficient mass or are capable of developing sufficient force to have dynamic characteristics of their own, which, because of the direct coupling with the vibration of the hull girder, can significantly influence the total or global pattern of vibration of the ship. In analyzing vibration patterns of such large complex structures it is necessary to identify the principal reason for excessive vibration, when observed. Although the excitation of the substructure generally originates at its attachment to the hull girder, excitation can come from machinery or active equipment mounted in the major substructure. Excessive vibration of a major substructure may result from structural resonances in the substructure or in the method of attachment of the substructure to the hull girder. Because of the mass involved and method of attachment, the major substructure can amplify the response of the hull girder.

To develop the optimum (simplest and least cost) solution to such problems it is necessary to conduct full-scale studies in sufficient depth to identify the primary problem. Structural analyses may or may not be required based on the complexity of the problem and the experience of the investigators.

Typical major substructures would include deckhouses; main deck structures; large propulsion machinery systems, particularly large slow diesels; and other heavy installations, including their foundations, such as boilers, reactors, large weapon systems, rudder, etc.

3.3.1 Response of Major Substructures

Figure 3-8 shows some possible modal patterns of vibration frequently found in aft deckhouse structures when excited by flexural and longitudinal vibration of the hull girder. Those shown indicate longitudinal vibration and include:

- Superstructure shear deflection
- Superstructure bending deflection
- Superstructure support deflection with rigid body motion
- Vertical hull girder vibration
- Longitudinal hull girder vibration

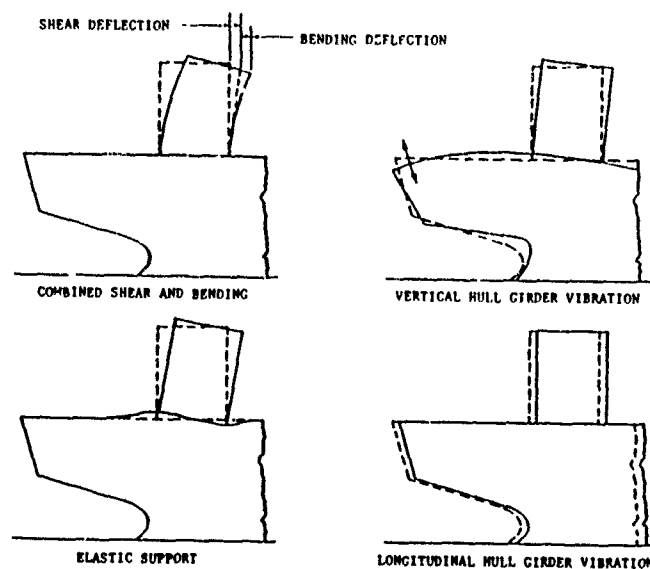


Figure 3-8. Superstructure Longitudinal Vibration

The dynamic response characteristic of the superstructure is primarily a function of superstructure shear stiffness and supporting structure vertical stiffness and the degree of coupling to hull girder modes. The superstructure rigid body motion is mostly due to hull girder response.

The resolution of structural problems of this type generally require investigators of considerable experience, a considerable test program, and dynamic analyses to obtain the optimum solution. Specific recommendations can only be applied to specific problems. In many cases, however, experienced investigators can radically simplify the approach.

3.4 VIBRATION OF LOCAL STRUCTURAL ELEMENTS

Local structural elements refers to minor structural assemblies, relative to major substructures previously referred to, and may be identified as panels, plates, girders, bulkheads, platforms, handrails, minor equipment foundations, etc. These "local structures" are components of larger structures (major substructures) or of the hull girder. Most problems encountered aboard ship occur in local structural elements and are the result of strong inputs received from the parent structure amplified by resonance effects in the local structure or responding to vibratory forces generated by mechanical equipment attached to the local structure. In some cases problems are generated by the improper attachment of shipboard equipment, even when the equipment has no self-exciting forces (passive equipment).

During the design of the ship, details of local structural elements and methods of installation of shipboard equipment are frequently based on practical experience and dynamic analyses are rarely performed. Although this approach is satisfactory in most cases, many problems arise or result from subsequent modifications. Most shipboard vibration problems fall in this category, and are generally amenable to easy and simple solutions once an understanding of the problem is obtained.

3.5

VIBRATION OF SHIPBOARD EQUIPMENT

Shipboard equipment is defined as all equipment installed aboard ship as a permanent part of the total ship system. It may contribute to the propulsion system, auxiliary, communication, control, or life support systems, and will include joiner work, furniture, etc. For convenience all such equipment is classified as "passive" or "active". In regard to vibration problems of shipboard equipment it is useful to separate the two.

3.5.1

Passive Shipboard Equipment

Passive equipment is all shipboard equipment permanently attached to the ship structure but which has no moving parts and/or produces no exciting forces. Typical examples would include heat exchangers, radio equipment, switchboards, joiner work, furniture, piping, etc. Excessive vibration of such equipment could be damaging to the equipment and adversely affect the operation of the unit or the system of which it is a part. In most cases, specific environmental limitations exist, whether identified or not. In some cases vibration limitations are established for shipboard equipment, particularly in naval equipment. At the present time international standards are under consideration for qualification of shipboard equipment for environmental vibration. Equipment which is sensitive to vibration, such as electronic equipment, is frequently installed on resilient mountings. A common difficulty arises from an improper selection of mountings.

In the evaluation of shipboard vibration, as it affects passive shipboard equipment, the same approach is recommended as is used for the vibration of local structural elements. The vibration encountered is normally associated with the response of the supporting structure and may be related to the main propulsion system, to the forces generated by nearby machinery, or, indeed, may be directly attached to a machine, such as a gage on a diesel engine. As in the previous case, the problem results from strong input forces and/or a resonant magnification of the attachment method or an internal mechanical resonance.

3.5.2

Active Shipboard Equipment

In contrast to the characteristics of passive shipboard equipment, active shipboard equipments (e.g., pumps, compressors, generators) have moving parts which frequently include sufficient mass to produce vibratory forces which, when combined with the dynamic characteristics of the supporting structure, would be capable of creating problems when operating. Such equipment support systems may also include resilient mountings which can reduce the transmission of self-generated forces to the supporting structures but which can also amplify the low frequency vibration generated by the ship's propulsion system.

The principal problems associated with the vibration of active shipboard equipment relates to the forces generated by the equipment itself and those transmitted to the equipment through the ship's structure. These forces can usually be distinguished by the different frequencies present. The supporting structure and the mounting system, if used, can generally be modified if necessary, without great difficulty.

3.6

VIBRATION OF MAIN PROPULSION MACHINERY

The main propulsion machinery includes all components from the engine up to and including the propeller, and thus contributes to the vibration of the ship and to dynamic stresses within the propulsion system itself by forces generated both by the propeller and by the propulsion system components. The propeller forces and their effect on hull vibration were discussed previously. In this section we will discuss dynamic forces generated by the propulsion system and the effect of these forces on the vibratory characteristics of the total propulsion system.

Of major concern is the dynamic stresses within the system and within the system components, and the control of dynamic forces generated by the propulsion system which contributes to the vibratory characteristics of the total ship. Although the vibration of both the ship's hull and main propulsion machinery are interrelated, it is convenient, both in preliminary design studies and in the control of shipboard vibration, to conduct independent studies on the propulsion system. It is necessary, however, to include actual or empirical factors related to the ship's structure which form an important part of the effective mass-elastic system under study, such as the stiffness of the thrust bearing foundation, when evaluating the response of longitudinal vibration of the propulsion system.

The main areas of concern, and which can give rise to troublesome vibration or dynamic stresses, include:

- Dynamic Unbalance and Misalignment
- Dynamic Shaft Stresses
- Longitudinal Vibration
- Torsional Vibration
- Lateral Vibration

The following sections will cover the above topics and include both the excitation and response of the propulsion system.

3.6.1 Dynamic Unbalance and Misalignment

Dynamic and/or hydrodynamic unbalance of the propeller, dynamic unbalance of shafting, bull gears, and other large components of the propulsion system operating at propeller-shaft speed may contribute to objectionable hull vibration, particularly if the exciting frequency falls in resonance with a natural frequency of the hull. Such difficulties may also arise from the primary (1st order) or secondary (2nd order) unbalanced forces in large, slow-speed diesel engines or from serious shaft misalignment (1st order).

It is generally true, however, that the vibration occurring at these low frequencies (1st or 2nd order) will be particularly objectionable to

humans when operating at the lower hull resonances. Vibration which exceeds the recommended criteria should be corrected to prevent local damage and/or bearing wear down. Specific corrective action may be required to control primary and secondary unbalances in slow-speed diesel engines.

Specific unbalance tolerances or machine vibration limits of high-speed components such as turbines and compressors are normally established by the manufacturer. When the vibration of such units exceed recommended criteria it may result in potentially dangerous problems with the equipment itself, or may cause resonances of local foundations, attached piping, or components. In the absence of manufacturers criteria, the criteria given in this guide should be used. Care should be exercised to distinguish between hull-excited and machine-excited vibration in order to properly determine corrective action required.

3.6.2 Dynamic Shaft Stresses

Propulsion shafting is normally designed in accordance with Classification Society Rules (A.B.S., Lloyds, etc.) and, in some instances, by Navy rules [3-6]. With normal design practice, periodic inspections, and proper maintenance procedures no difficulty should be experienced with propulsion shafting during the life of the ship. However, experience has indicated serious difficulties, including shaft failure, could result during normal operating conditions [7,8].

Shaft problems are related to dynamic stresses in most cases, magnified by corrosion fatigue. Such problems may be caused by the eccentric thrust, precipitated by adverse flow conditions at the propeller, and aggravated by misalignment and/or faulty shaft seals. Excessive stresses associated with torsional vibration in slow-speed diesel engine drives is also a potential problem area.

As a minimum, the complete propulsion system should be evaluated for acceptable steady and dynamic stress levels during the design phase, and verified during ship trials. Maintenance procedures should check for corrosion and fatigue cracks at the propeller keyway and at the shaft at the forward end of the propeller hub. Bearing wear and wear of shaft seals should also be checked.

3.6.3 Longitudinal Vibration

The propulsion system may exhibit excessive longitudinal vibration caused by alternating thrust generated by the propeller at blade frequency or harmonics of blade frequency. The vibration is considered excessive if it exceeds the criteria given for machinery vibration, and can be particularly damaging to thrust bearings and/or reduction gears. Depending on structural characteristics, the alternating thrust forces transmitted to the ship through the thrust bearing can result in serious local vibrations in the engine room and to serious superstructure fore and aft response. Figure 3-9 shows the longitudinal vibration of a typical propulsion shaft. The addition of the main engines and reduction gears to the mass-elastic system is required for complete system

evaluation. The forces transmitted to the ship's structure are primarily dependent on the total mass of the system shown in Figure 3-9 and the combined thrust bearing and foundation stiffness.

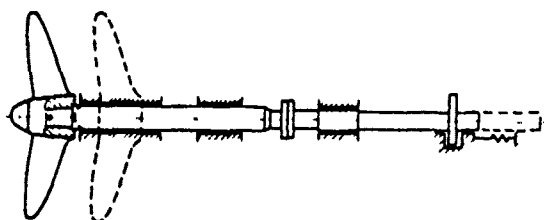


Figure 3-9. Longitudinal Vibration of Shafting

In general, longitudinal and torsional vibrations of propulsion systems may be considered as independent of one another, but this is not always the case. The propeller couples the longitudinal and torsional degrees of freedom of the system to some extent under all conditions, but the coupling effect is significant chiefly when the critical frequencies that would exist without this coupling effect are close to one another. In such cases the mode excited is actually a longitudinal-torsional mode and the excitation involves a generalized force which includes both torque and thrust variations. This phenomenon is of particular concern in diesel drive systems.

While longitudinal vibration may be observed aboard ship, to properly instrument and evaluate against the various criteria will require a dynamic analysis for correlation purposes and, in most cases, further analyses to determine optimum corrective action. Vibration specialists should be obtained for such problems and for total system evaluation during ship trials.

3.6.4 Torsional Vibration

Torsional vibration of the propulsion system may be excited by the alternating torque produced by the propeller and/or the engine harmonics in a diesel drive system. Ordinarily torsional resonances within the shafting system shown in Figure 3-10 does not produce serious vibration problems in the ship's structure although it can produce damaging effects in reduction gear drives, particularly under adverse sea conditions. In diesel engine drive system of all types, torque reactions can be a major concern in structural vibration and torsional resonances can be damaging to system components.

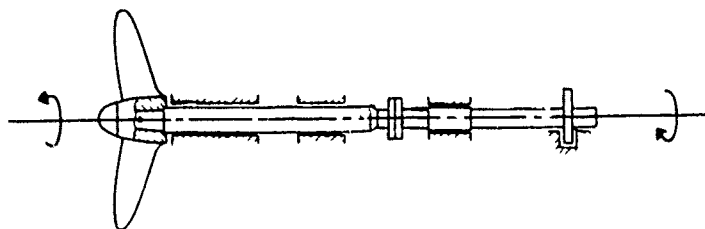


Figure 3-10. Torsional Vibration of Typical Shafting System

Although the evaluation of torsional vibration of the shafting is subject to classification rule requirements, it is also considered necessary to carry out a torsional vibration analysis of the complete propulsion system in the design phase and verify the system response characteristics during ship trials. As in the case of longitudinal vibration studies, experienced personnel are considered necessary for the evaluation and resolution of shipboard problems. For more detailed information on the subject see "Practical Solutions of Torsional Vibration Problems" [3-9] and "BICERA" [3-10].

3.6.5 Lateral Vibration

The propulsion shaft system, Figure 3-11, is normally designed so that the fundamental lateral or whirling critical speed is well above the running speed. Background information and calculation procedures are given by Jasper [3-11], Panagopulos [3-12], and Navy Design Procedures [3-6]. The fundamental mode of vibration is referred to as "forward whirl", is excited by mass unbalance, and at resonance poses a serious danger to the propeller-shaft system. The frequency of the system is significantly influenced by the effective point of support of the aft bearing and the stiffness of the bearing supports. Figure 3-12, taken from Det Norske Veritas Guidelines [3-5] shows the influence of the position of the aft bearing support on the frequency of the whirling critical.

Misalignment or serious bearing wear down can result in high dynamic stresses in the shaft, dynamic magnification of bearing reactions and increased hull vibration, overheating, and increased bearing wear down. On the assumption that the design was satisfactory initially, good maintenance is required to keep it that way. The use of roller bearings or self aligning bearings, and attention to dynamic balance will minimize potential problems.

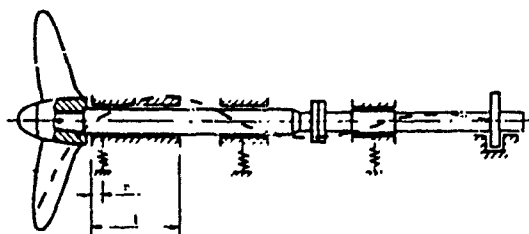


Figure 3-11. Whirling Vibration of Shafting

MAJOR WHIRLING CRITICAL

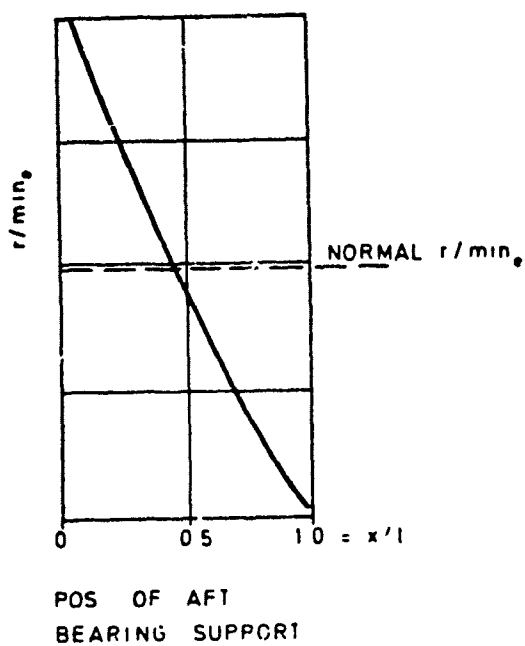


Figure 3-12. Position of Aft Bearing Support

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SECTION 4.0

MEASUREMENT METHODS

This section deals with the equipment, techniques, and procedures used to obtain and reduce shipboard vibration data. The evaluation of the levels of vibration is covered in Section 8.

A limited amount of information can be obtained without equipment, just from feeling the vibration. For example, the frequencies of the lower hull modes, and shaft rotation can sometimes be determined by counting the cycles during a known period of time. Often blade frequency can be detected even though it is normally too fast to count. A severe cavitation of the propeller can be identified by the noise in the stern area, which sounds like a loud random hammering on the hull. An object, such as a coin placed on a vibrating machine, will rattle if the vertical vibration level is over one g. Critical speeds of machinery or equipment can often be observed visually, or detected by the noise it produces. Such observations, especially by experienced ship operators or engineers, are valuable, and may be sufficient in evaluating simple problems. However, it is easy to be deceived, and if there is any doubt, instruments should be used.

4.1 INSTRUMENTATION

This section will provide the information on mechanical instruments, vibration transducers, signal conditioning and recording equipment that would have to be considered in planning shipboard vibration measurements. It treats the various types of equipment in general terms, so when specific equipment is selected, reference will have to be made to the manufacturer's instructions for the actual operation of the equipment.

4.1.1 Mechanical Instruments

There have been many mechanical and optical instruments developed for measuring vibration, most of them before electronic means were highly developed. Two of the most common are still used and are briefly described.

4.1.1.1 Reeds

These instruments consist of metal reeds attached to a case which is held hard against a vibrating object. If the natural frequency of the reed coincides with the frequency of vibration, the reed's amplitude will be magnified to the point where it can be seen. One type of reed instrument (Frahm reeds) has a number of reeds of different lengths (and frequencies) attached to the same case, and the reed with the highest amplitude is closest to the frequency of vibration. Other instruments use a tunable reed where the length can be adjusted. The amplitude of the reed can be calibrated to give the amplitude of vibration (Westinghouse Reed Vibrometer).

4.1.1.2 Askania

This is a hand-held instrument with a probe that is pressed against a vibrating object. The relative displacement between the probe and the casing is mechanically amplified and displayed on a 1-inch wide strip of paper as a time history, from which the amplitudes and frequencies of vibration can be determined. Low frequency signals are often unreliable due to the difficulty in holding the instrument still.

4.1.2 Transducers

Several of the basic types of transducers used for vibration measurements are discussed. The theory of operation is intentionally very brief, but the features that must be considered in their application are covered in more detail. The type of transducers chosen will depend on what type of information is required.

4.1.2.1 Accelerometers

The most common type of accelerometer is the piezoelectric type. This consists of a mass mounted on a crystal which generates an electrical charge proportional to the acceleration of the mass. It is generally small in size and models can be found that can measure from 1 Hz to 10 kHz. They require a charge amplifier and the cables from the transducers to the amplifiers must be the coaxial type. If the cables between the transducers and amplifiers are long and/or are subject to vibration, a charge converter should be inserted in the cable a few feet from the transducer. A two-conductor shielded cable can be used between the charge converter and the amplifier.

If frequencies down to D.C. must be measured, a strain gage or piezoresistive accelerometer can be used. The strain gage type has the mass mounted to a flexural or other support which bends or deflects when the accelerometer vibrates. Strain gages are used on the support to measure the deflections. The piezoresistive types have the mass mounted to piezoresistive elements which act much as strain gages do. These must be used in conjunction with strain gage amplifiers. Four conductor shielded cables must be used between the gages and the amplifiers.

Another type of accelerometer is the servo accelerometer. This type contains a seismic mass with a coil of wire around it. Motion of the mass with respect to the case due to acceleration is detected and a current imposed on the coil to keep the mass moving with the case. The amount of the current is the signal output. The frequency response of one brand of this type is from D.C. to approximately 500 Hz. It has a built-in amplifier, and requires a D.C. power supply.

Acceleration signals are suitable for analyzing many types of machinery problems where high frequencies are of interest such as bearing wear, because the high frequencies are accentuated. However, if lower frequencies are of interest, such as a once per revolution signal, that signal may be lost in a

multitude of high frequency signals due to bearings, flow noise, etc. In the analysis section some techniques will be discussed to overcome this problem.

4.1.2.2 Velocity Gages

A velocity gage is constructed with a magnetic core, mounted on springs, that oscillates within a coil generating a signal which is proportional to velocity. It generally has an internal resonant frequency of 5 or 6 Hz, is highly damped to eliminate a high output at the resonance, and has a useful frequency range of about 4 to 600 Hz. The lower frequencies require a correction, but above approximately 10 Hz the output is essentially linear. Two conductor shielded cables are sufficient for velocity gages. The velocity signal, and alternatively, the displacement signal if it is integrated, is usually the most appropriate quantity to measure for ship vibration problems if high and very low frequencies are not of interest.

4.1.2.3 Displacement Gages

The most common type of gage measuring displacement directly is the non-contact proximity probe. It generally consists of a proximator and a probe. The proximator generates a high frequency signal which produces a magnetic field around the probe. The closeness of a metal "target" alters the eddy currents, modulating the high frequency signal which is demodulated by the proximator producing a signal which is proportional to the distance between the probe and the target. The frequency response is from D.C. to several kHz depending on the frequency of the carrier. The proximator requires a D.C. power supply. The cable between the proximator and the recorder or signal conditioner can be two conductor shielded.

The proximity probe is used extensively in rotating machinery studies, measuring the relative displacement between the stationary and rotating parts. If the actual displacement of a rotating body is required, the displacement of the casing or whatever the probe is mounted to must be measured by some other means. The most severe limitation of these is that they measure only a range of a few to 50 or 100 mils displacement unless the probe is quite large.

There are other displacement measuring transducers that have each end mounted to objects to measure the relative distance between them. Two such devices are the Linear Variable Displacement Transducer (LVDT) and the Linear Motion Transducer (LMT).

4.1.2.4 Strain Gages

Strain gages are resistive elements that change their resistance when they are stretched or compressed. They are mounted with an adhesive to the object and in the direction in which strain is to be measured. The change in resistance is small so the measurement is usually made with the gage electrically connected in a Wheatstone bridge.

A D.C. power supply of 5 to 10 volts is required. Often this is built into strain gage amplifiers.

If the strain in a shaft is being measured, telemetry equipment or slip rings must be used. The arrangements of the gages for measuring axial, torsional, and bending strains in the shaft are all different, and must be precisely positioned.

The techniques for installation, calibration, and measurement are quite involved, and require experienced personnel.

4.1.3 Signal Conditioners

Signal conditioning equipment is used to amplify signals to a level where they can be conveniently recorded or displayed. Depending on the application, they may also include power supplies for the transducers, filters, calibration features, integrators, and parts of Wheatstone bridge circuits.

4.1.3.1 Differential Amplifiers

Differential amplifiers in their simplest form use voltage inputs and merely amplify them to a level suitable for recording. The most common additional feature found on these is an internal calibration signal which is fed into the amplifier. This could be a sinusoidal signal or a D.C. step of known value. This is convenient for calibration of the amplifier and recording equipment, but does not calibrate the sensitivity of the transducer. That would have to be done by other means or provided by the transducer manufacturer.

The output of this or any type of amplifier is usually a voltage which can be used as input to high impedance devices only. This would include tape recorders, many strip chart recorders, spectral analyzers, oscilloscopes, etc. The most common low impedance device used is an oscillograph that uses galvanometers. These are designed such that the display is proportional to the input current rather than voltage. Special "galvo outputs" are required on the amplifiers to drive these devices.

4.1.3.2 Charge Amplifiers

Charge amplifiers are used with piezoelectric accelerometers and they convert the charge generated by the accelerometer to a voltage proportional to the acceleration. Some of these have a feature which allows the use of remote charge converters, in which case the amplifiers become basically differential amplifiers. In addition, however, they provide a D.C. power supply superimposed on the signal cables to power the remote charge converters. If the remote charge converters are used it is possible to use a separate power supply and differential amplifiers provided the charge converters have the appropriate connectors.

Since acceleration signals often have high level high frequency signals which are not of interest, charge amplifiers often have built-in filters to limit the amplifier output to 50, 100, 500 Hz, etc. If velocity or displacement is desired, some amplifiers can integrate the signal once to get velocity, or twice to get displacement. If this feature is used, the characteristics of the

Integrators at low frequencies should be checked. Often the transducer can be used at lower frequencies than the integrators.

Charge amplifiers, like differential amplifiers, may have internal calibration signals and/or galvanometer outputs as well as voltage outputs.

4.1.3.3 Strain Gage Amplifiers

A strain gage amplifier is basically a differential amplifier, but usually includes most of the following features. It should have a D.C. power supply rated at up to 10 volts. It usually has a shunt calibration switch which places a calibration resistor across one arm of the bridge. This changes the resistance of that arm to correspond to a known strain.

The strain gage amplifier also has a bridge balance network which adjusts the voltage across adjacent arms to zero the output when the structure being measured is not under load.

Again it is recommended that only experienced personnel be used for strain measurements.

4.1.3.4 Filters

The most common type of filter used in ship vibration measurements is a low pass filter which passes all frequencies below a set frequency and blocks all those above it. As mentioned before, this can be used to eliminate high frequencies such as from bearings, flow, etc. The "cut-off" is not sharp, however, and the filter characteristics should be considered when using them.

A high pass filter can be used to eliminate unwanted low frequencies such as ship motion, or low frequency hull vibration. Again the cutoff is not sharp.

Both high pass and low pass filters can be used on the same signals, either as a "band-pass" filter or as a "band-reject" filter. The band pass filter would be used if there are both low frequency and high frequency signals that are unwanted. Also it might be used to isolate a certain component, such as blade frequency, in which case a narrow band would be used. More will be said about filters in the analysis section.

4.1.4 Recorders

This section discusses four means of recording or observing vibration data: meters, oscilloscopes, oscillographs, and tape recorders. Often data is fed directly into a frequency analyzer and the peaks recorded by hand or the entire plot put on hard copy. Although this can be considered a type of recording, frequency analyzers are discussed under Section 4.3, Analysis and Reporting of Data.

4.1.4.1 Meters

This is the simplest type of display used for vibration data. The amplitude of the A.C. component of a signal is displayed on a meter. If there are no filters in the signal conditioning system, the meter displays an "overall" level of vibration and no indication of the frequencies involved. This is useful for many machinery monitoring applications where a change in overall vibration level is usually indicative of a problem. Sometimes enough is known about the machine that the frequency can be assumed. More often some type of frequency detection is used for diagnostic purposes after a problem has been detected.

A filter can be used in conjunction with a meter to obtain the frequencies and amplitudes of many of the components in the signal. Normally a narrow band filter is used in this manner and is tuned manually. This procedure is severely limited by the gradual cut-off of most analog filters. This results in lower level components being masked by the higher level signals if they are close in frequency. "Close in frequency" will be defined differently for different filters, but could be a factor of two or three. Also two components that are very close are difficult to distinguish. Nevertheless, this procedure is very useful for many applications that have a limited number of frequency components.

A meter, be it with or without a filter, is used mostly with machinery where the amplitudes do not modulate significantly. Most hull, superstructure, and main shaft vibration measurements exhibit a modulation such that the maximum amplitudes are two to five times the average. This modulation varies with the quantity being measured, sea state, and other factors. At any rate, it is difficult to deal with using a meter because the needle fluctuates too much to get a good reading. Also, the modulating signal is a statistical quantity and a sufficient number of cycles must be observed. This is impossible with a meter.

Many packages are available from manufacturers, such as IRD, that include a transducer, tunable filter, and calibrated meter.

4.1.4.2 Oscilloscopes

Oscilloscopes display a time history of the vibration signal on a screen as it occurs. In many scopes a segment of the signal can be "captured" and retained for examination. The amplitude of the signal can be read from the screen if it is not modulating too much. If there is a dominant frequency component, the frequency can be obtained from the screen as well, by observing the period on the x-axis. If there is more than one frequency component, getting frequency information from a scope is difficult.

Oscilloscopes are more useful as monitoring devices to see if a reasonable looking signal is being recorded, and to observe the general characteristics of the signals. For multiple channels it should be used with a switchbox so all channels can be checked. Precise measurements of the amplitudes and frequencies should be done by other means. See Section 4.3.

4.1.4.3 Oscillographs

Oscillographs produce a hard copy of the time history of a number of vibration signals simultaneously, and these permanent records can be analyzed any time after the data is taken. The procedures for obtaining the amplitudes and frequencies from these records are given in Manley (4-1), and briefly discussed in Section 4.3.1.

Where there are several modulating frequency components in a single signal, it becomes difficult and time consuming to analyze. Oscillographs are often used in preliminary analyses, and to get a feel for what is happening during the trial. It can also serve as a backup in case something should happen to a tape recorder used in parallel.

The distinction between "oscillographs" and "strip chart recorders" is somewhat vague. In general a strip chart recorder has a pen, perhaps thermal or ink, and is limited to a frequency response of 100 Hz or less. An oscillograph usually refers to a recorder that uses light beams directed towards a roll of light sensitive paper. Its frequency response could be as high as 5 kHz. The light beams might be controlled by galvanometers in which case a small mirror reflects a light source in response to the incoming signal. A later innovation uses fiberoptics to direct the light.

Galvanometers have low input impedance, and are current controlled devices. Signal conditioning equipment used with galvanometers must have an output terminal designed specifically for them. The fiberoptics equipment and strip chart recorders normally have differential amplifiers built in with gain controls, so that the same signal can be used for the oscillograph or strip chart recorder as is used for the tape recorder.

4.1.4.4 Tape Recorders

The use of a tape recorder provides a versatility that cannot be obtained by the other methods discussed. It reproduces the electrical signal which can then be analyzed by any method desired. This flexibility is desirable because the requirements for analysis cannot always be totally predicted before the trial. The procedures involved are fairly simple, but must be rigorously observed.

The level of the input signals must be constantly observed and controlled by the signal conditioning equipment so that the signal is as large as possible to reduce the signal/noise ratio, but not too large for the tape recorder, so the signal is not clipped. The level can often be monitored with a meter built into the tape recorder. An alternative method is to monitor the level on an oscilloscope.

An oscilloscope is desirable anyway to judge whether the signals look reasonable or not. Usually intermittent connections, 60 Hz noise, or other problems can be instantly detected. The scope should be used also to monitor the tape's reproduced signals AS THEY ARE BEING RECORDED. This will require extra wiring and switching arrangements, but it is imperative that the trial

engineer be able to routinely check if all signals are the same when they are reproduced as they were when recorded.

The tape speed selected will depend on the frequency response of the signals to be recorded, and whether the tape recorder uses "wide band" or "standard band". Check the instructions for the recorder to select tape speed.

4.2 QUANTITIES TO BE MEASURED

The first choice to be made regarding the type of data to be recorded is between displacement, velocity, and acceleration. In addition, the appropriate frequency range should be known in advance. Although most ship vibration data will be recorded in the form of a time history, sometimes the data will be fed directly to a frequency analyzer or other analysis equipment and a hard copy of the frequency spectra or some other type of plot obtained. In some very simple applications, data may be read from the equipment and recorded manually as the trial is conducted.

This section discusses the measured quantities in general. Section 4.4 (Transducer Locations) makes specific recommendations for different situations.

4.2.1 Displacement, Velocity, or Acceleration

In most cases ship vibration data will be analyzed in terms of its frequency components, so the characteristics of sinusoidal motion should be considered. These were discussed in Appendix A, Section A.2, Vibration Fundamentals, where it was shown that the displacement, velocity, and acceleration amplitudes of sinusoids differ only by factors of the frequency, with the higher frequencies being accentuated by acceleration, the lower by displacement. Vibration engineers often consider displacement the appropriate quantity to observe for machines that operate below 1000 rpm, velocity for 1000 to 10,000 rpm, and acceleration for those above 10,000 rpm. This is assuming they are interested in things such as unbalance, misalignment, etc. If a high frequency, such as a bearing frequency is of interest, even on a low speed machine, acceleration may be the best quantity to observe.

Another factor to consider in choosing displacement, velocity, or acceleration is what is critical in determining damage to the machine or structure. Displacements will normally be proportional to stresses in a structure and can be compared to known clearances, etc. Velocity gives an indication of the energy dissipated through vibration, which is often a good indicator of damage to a machine. Human comfort level is more closely related to velocity than displacement or acceleration. Acceleration is normally proportional to the forces applied to the vibrating object.

The final choice is often determined by the characteristics of the transducers available, particularly the frequency range of the transducer. See the section on transducers for details.

4.2.2 Frequency Range

The frequency ranges of transducers, signal conditioners, and recording equipment must be chosen to match the frequency components of interest in the data being recorded. Also, frequencies known to be present but not of interest are often excluded. Ship motion (roll, pitch, etc.) usually falls below 1 Hz. Hull girder modes and those of major structures, such as deckhouses, masts, etc. may be from 1 to 10 Hz for the lower modes. Propeller shaft rotational, main engine rotational, gear frequencies and blade rate can all be determined from the machinery characteristics.

Even though unwanted frequencies can be filtered out in the analysis of the data, it may be a good idea not to record them in the first place because the level of the frequency components of interest will be harder to separate from the "noise" later. Also, in the case of frequency analyzers, choosing a broader frequency range than necessary will decrease the resolution and accuracy of the results.

4.2.3 Time History or Frequency Spectra

Often it is tempting to obtain frequency spectra directly rather than go through the step of recording on tape and then later obtaining the spectra. This is appropriate for rotating machinery where the vibration is self-excited. In this case the amplitudes are usually fairly constant and a large sample is not required. Most vibration data routinely gathered from ships, however, are propeller excited, or excited by the sea, or some combination, such as hull girder vibration. For this type of data, the amplitudes are modulated and have a randomness associated with them. It requires a sample large enough to account for the randomness, which for blade frequencies and its harmonics, should be two or three minutes. If multiple channels are being recorded, as is usually the case, time will not permit "on-line" analysis during the trials.

4.3 ANALYSIS AND REPORTING OF DATA

Traditionally, analysis of shipboard vibration records has been performed using the methods described in "Waveform Analysis" by R.G. Manley (Reference 4-1). The various schemes proposed in that work allow one to extract maximum values and frequencies of the components of a complex waveform. If there are several components present "manual" analysis is time consuming and requires good judgment (i.e., experience). Common practice has been to analyze a record of one to several minutes duration and report the "maximum repetitive" amplitudes of the predominant components. This practice was established because the maximum values are those responsible for discomfort and structural damage.

Recently, very fast digital electronic analyzers have become available at a reasonable cost. These analyzers offer several advantages over manual analysis, primarily speed and repeatability. No judgment is required by the machine. Unfortunately, the machines do not look at data the same way as experienced vibration engineers. Most of these machines perform a discrete fourier transform on an electrical analog of the vibration waveform. The analysis is accomplished by an efficient algorithm known as a Fast Fourier Transform. The

result of this transform is the Root Mean Square amplitude of each component within the analysis range. For a pure sinusoid the ratio of the peak amplitude to the RMS amplitude is $\sqrt{2}$. In order to obtain the maximum repetitive values, the RMS values must be multiplied by "crest factors" which include both the $\sqrt{2}$ factor and an amplitude modulation factor. There are limited data on crest factors at present, but they are known to vary with sea state and location of measurement, at least. Normally they are in the range of 2 to 4, but in adverse weather, values as high as 6 to 9 have been reported. The International Standards Organization (ISO) recommends that a factor of 2.5 be used unless there is enough data available to establish a more appropriate factor.

Six methods of analysis are discussed below. Not enough research has been done to evaluate these methods in relation to each other, but each has its own advantages, and can be used to compare quantities analyzed by similar methods.

4.3.1 Manual Method

The "manual" method of analysis involves measuring frequencies and amplitudes of vibration components on an oscillograph record. The subject is covered in great detail in Manley, Reference 4-1, and only a few basics are covered here.

The analysis of waveforms is based on the principle that any periodic record is a superposition of sinusoids having frequencies that are integral multiples of the lowest frequency present. The lowest frequency is determined by the smallest portion of the record that repeats itself, or one cycle. Figure 4-1 shows several waveforms and indicates the extent of one cycle, some of which are not obvious.

The first trace, (a), is essentially a sinusoid with a constant amplitude. The double amplitude of vibration is obtained by measuring the double amplitude of the trace as shown, and multiplying by the sensitivity of the measuring/recording system which is found by calibration. The frequency is found by counting the number of cycles in a known time period. The time on oscillographs is indicated by timing lines (a convenient rate for many shipboard applications is 10 lines/sec.) or simply by knowing paper speed. For trace (a), the frequency is 6 Hz. Accuracy is improved if the number of cycles in a longer section of record is used.

Trace (b) is the superposition of two sinusoids with one cycle of the lowest frequency shown. The components can be separated by drawing sinusoidal "envelopes" (upper and lower limits) through all the peaks and troughs as shown. The amplitude and frequency of the low frequency component is that of the envelope (frequency is about 2.7 Hz). The vertical distance between envelopes indicates the amplitude of the high frequency component, and the high frequency (about 8 Hz) can usually be counted. In this example the frequencies differ by a factor of three.

Often signals look like trace (c), where the envelopes are out of phase, causing "bulges" and "waists". This signal is caused by two components

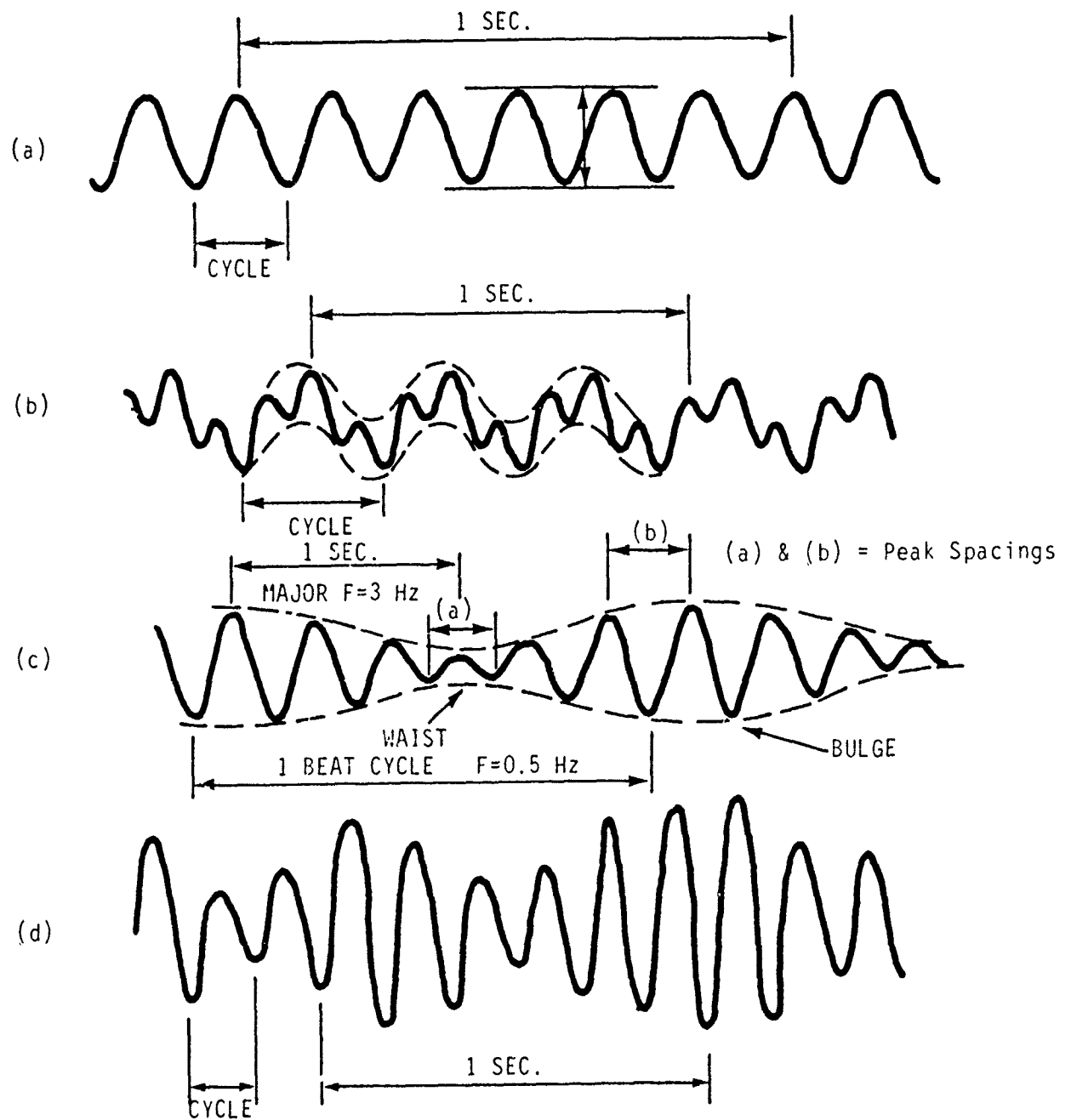


Figure 4-1. Various Types of Waveforms

that are close in frequency, and is called "beating". The peaks of the two signals alternately add and subtract. Other characteristics of beating are that the lengths of the beats are about the same, and the spacing between the peaks at the bulges is different from that at the waists. The amplitudes between the envelopes at the bulges and waists represent the sum and difference respectively of the components. Thus, if the components' amplitudes are x_m for the major, and x_n for the minor, measurements show that:

$$\begin{aligned} x_m + x_n &= .7'' \\ x_m - x_n &= .2'' \end{aligned}$$

Solving simultaneously by adding:

$$2x_m = .9'' \text{ or } x_m = .45'', \text{ so } x_n = .25''$$

These record amplitudes must be multiplied by the system sensitivity to get actual amplitudes. The major frequency can be found by counting the number of peaks as before. In trace (c) it is 3 Hz. This frequency is also some integral multiple of the beat frequency, in this case 6 times. The frequency of the minor component is either one more (7) or one less (5) times the beat frequency. The spacing of peaks at the waist indicates which, since it reflects the major component. In trace (c) the spacing is closer so the major component has the higher frequency. If the spacing were farther apart, the major component would have the lower frequency. In our example, the beat frequency is 0.5 Hz, the minor frequency is 5 times that, or 2.5 Hz.

Trace (d) shows a characteristic of most hull and propeller excited vibration on board ships. It looks similar to beating, but is actually only one component whose amplitude is varying (modulating) in response to wave action and flow variations into the propeller. This is distinguishable from beating because the length of the bulges are not likely to be the same, and the spacing of the peaks is the same at the bulges and the waists. For such records, the maximum repetitive amplitude is usually desired, which would be obtained from the highest bulges.

Unfortunately, many ship vibration records involve more than two components, will almost certainly involve modulation, and may be beating as well. One technique that saves a lot of time is to find sections of the record in which one component is temporarily dominant.

4.3.2 Envelope Method

This method of analysis involves filtering the signals to get the frequency component of interest and recording the result on a chart recorder at slow speed. This condenses a several minute record into an envelope just several inches long. The maximum repetitive value (MRV) can be immediately obtained visually.

Care must be taken that the filter used does not pass a significant amount of any component other than the component of interest. As an example,

using most analog filters to obtain blade frequency (frequency at which propeller blades pass a fixed point) is normally acceptable because the amplitude of the second harmonic is usually much lower than that of blade frequency. However, when analyzing the 2 x blade component, too much of the blade and the 3 x blade would pass to obtain meaningful results.

To illustrate how these errors can be anticipated, consider a Krohn-Hite Model 3550 Variable Filter. It uses a fourth order Butterworth function. The gain is given by Figure 4-2 and the following:

$$G_L = \frac{1}{\sqrt{1 + s^8}}$$

$$G_H = \frac{s^4}{\sqrt{1 + s^8}}$$

$$s = \frac{f}{f_o}$$

where G_L = Gain of low pass filter
 G_H = Gain of high pass filter
 f = Frequency
 f_o = Cut-off frequency setting

The effect that this filter will have on blade frequency and its harmonics can be illustrated by calculating the attenuations for various frequency ratios, S . The results are given in Table 4-1. It was assumed that the high pass and low pass filter settings were at 80 percent and 120 percent, respectively, of the frequency being passed.

From the table it can be seen that using the filter attenuates the wanted signal to 83.3 percent. Any results so obtained should be divided by .833 to get true amplitudes. To see how this step affects all the components, the entire table is divided by .833 to find the "normalized" attenuations.

The normalized attenuations show that if blade frequency is being filtered, 15.4 percent of the second harmonic also passes, an acceptable error if the second harmonic is significantly less than blade frequency, which is the usual case. When analyzing the 2 x blade component, 18.1 percent of the blade, and 45.4 percent of the 3 x blade passes, an unacceptable error.

To reduce this type of error, two filters can be used in series (Table 4-2). When analyzing 2 x blade, two filters result in 69.4 percent of 2 x blade, 2.3 percent of blade, and 14.3 percent of 3 x blade being passed. To normalize, divide by .694. The normalized attenuations show 3.3 percent of

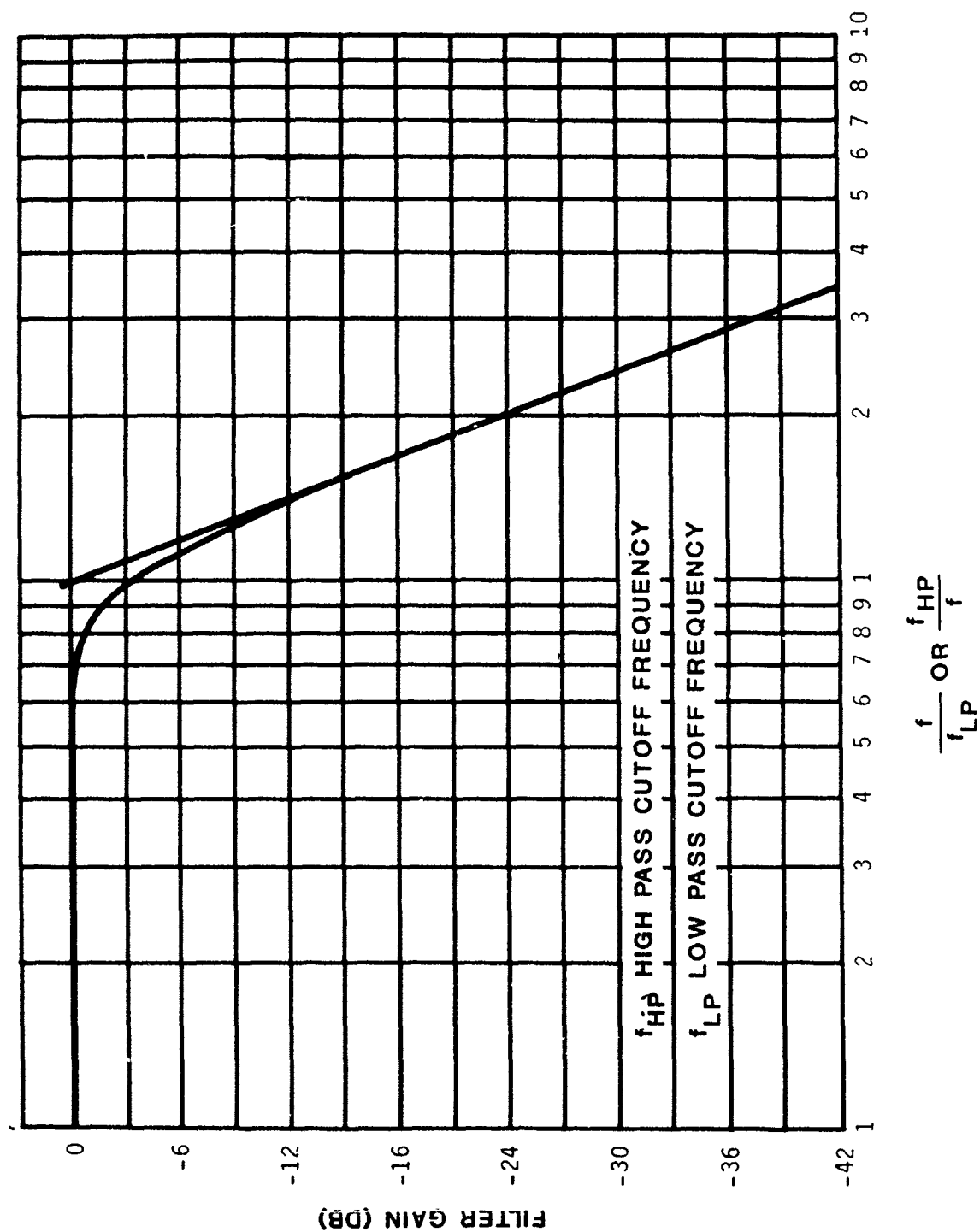


Figure 4-2. Normalized Attenuation Characteristics of Krohn-Hite 3550 Filters

Table 4-1. Calculated Attenuations Due to One Filter

| <u>FILTER SETTING</u> | <u>Actual Attenuations</u> | | | | <u>Normalized Attenuations</u> | | | |
|---------------------------|----------------------------|------------------|------------------|------------------|--------------------------------|------------------|------------------|------------------|
| | <u>BLADE</u> | <u>2 x BLADE</u> | <u>3 x BLADE</u> | <u>4 x BLADE</u> | <u>BLADE</u> | <u>2 x BLADE</u> | <u>3 x BLADE</u> | <u>4 x BLADE</u> |
| BLADE | .833 | .128 | .026 | .008 | 1.000 | .154 | .031 | .010 |
| 2 x BLADE | .151 | .833 | .378 | .128 | .181 | 1.000 | .454 | .154 |
| 3 x BLADE | .030 | .432 | .833 | .544 | .036 | .519 | 1.000 | .653 |
| 4 x BLADE | .009 | .151 | .604 | .833 | .011 | .181 | .725 | 1.000 |

Table 4-2. Calculated Attenuations Due to Two Filters

| <u>FILTER SETTING</u> | <u>Actual Attenuations</u> | | | | <u>Normalized Attenuations</u> | | | |
|---------------------------|----------------------------|------------------|------------------|------------------|--------------------------------|------------------|------------------|------------------|
| | <u>BLADE</u> | <u>2 x BLADE</u> | <u>3 x BLADE</u> | <u>4 x BLADE</u> | <u>BLADE</u> | <u>2 x BLADE</u> | <u>3 x BLADE</u> | <u>4 x BLADE</u> |
| BLADE | .694 | .016 | .001 | .000 | 1.000 | .024 | .001 | .000 |
| 2 x BLADE | .023 | .694 | .143 | .016 | .033 | 1.000 | .206 | .024 |
| 3 x BLADE | .001 | .187 | .694 | .296 | .001 | .269 | 1.000 | .426 |
| 4 x BLADE | .000 | .023 | .365 | .694 | .000 | .033 | .526 | 1.000 |

blade and 20.6 percent of 3 x blade is passed. This technique should be acceptable unless the 3 x blade component is unusually high in magnitude. Even two filters may not have a sharp enough cut-off to isolate the 3 x blade or higher harmonics.

For normal ship vibration signals, the following steps are recommended:

1. Analyze blade frequency with one filter and normalize results.
2. Analyze 2 x blade frequency with two filters and normalize results.
3. Subtract 15.4 percent of the 2 x blade amplitude from the blade frequency amplitude.
4. Subtract 3.3 percent of the blade frequency amplitude from the 2 x blade amplitude.

NOTE: The above percentages will vary with different filters.

In order to visualize the relationship between a normal oscillograph record of ship vibration and a condensed envelope, a 1-minute sample of filtered blade frequency vibration of a ship's stern was recorded at different speeds. Figure 4-3 shows the record at 25 mm/sec. Figure 4-4 shows the same record at successively slower speeds. At 1 mm/sec. the maximum repetitive value is conveniently read.

It is felt that this is presently the best method to obtain the MRV for blade and 2 x blade frequency from ship vibration records.

4.3.3 Spectral Method

Two types of spectra can be found with most analyzers: the common "average" and the "peak" amplitude. For both types the record is broken down into segments for frequency analysis. The segments vary in length with the frequency range of the analysis, but for ship vibration studies they would be several seconds long. For each segment, the analyzer finds the RMS level of each frequency component (i.e., in each frequency interval, or for each "line"). The number of lines (resolution) varies with the analyzer, but most often falls between 100 and 1000. Ship vibration records will normally be one or several minutes long and will contain many segments.

If the "average" spectrum is desired, the analyzer will average all the RMS levels found in like frequency intervals. The term "number of averages" is often used referring to the number of segments in each average, although the terminology is sometimes confusing. This report will refer to the values obtained with "average" spectra as "average RMS" values.

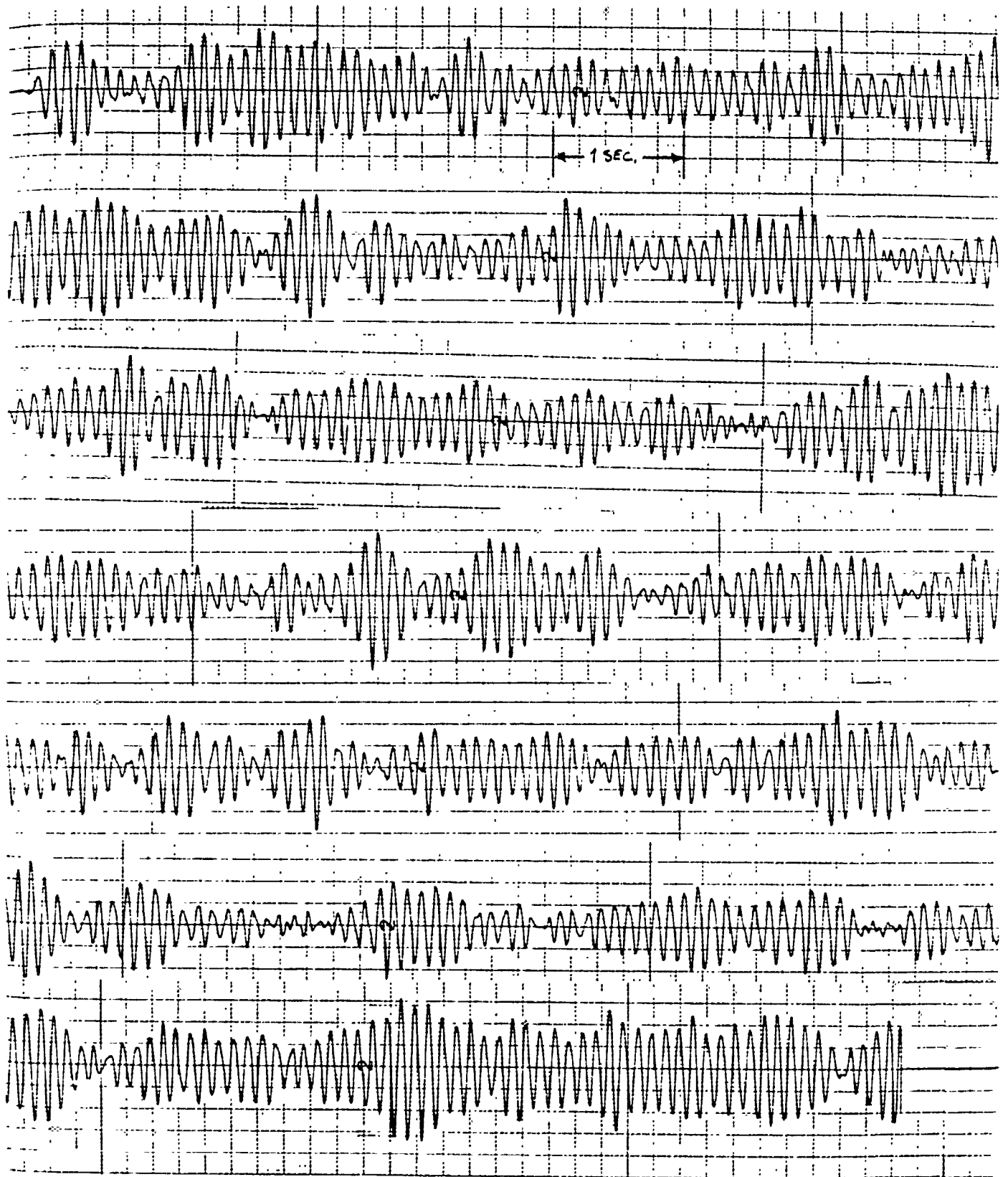
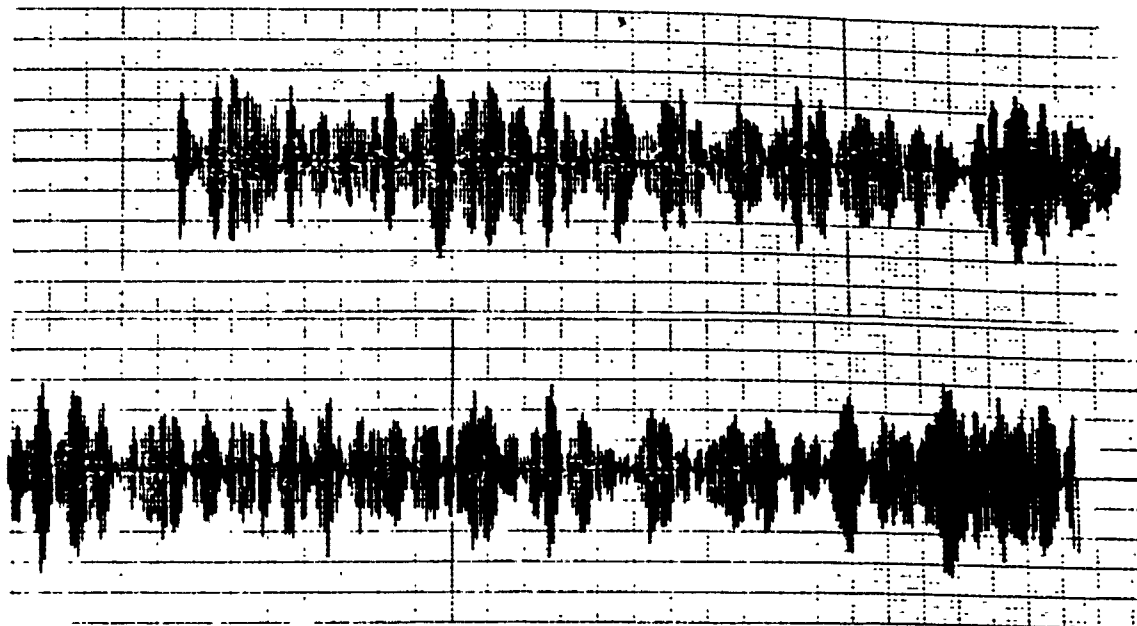
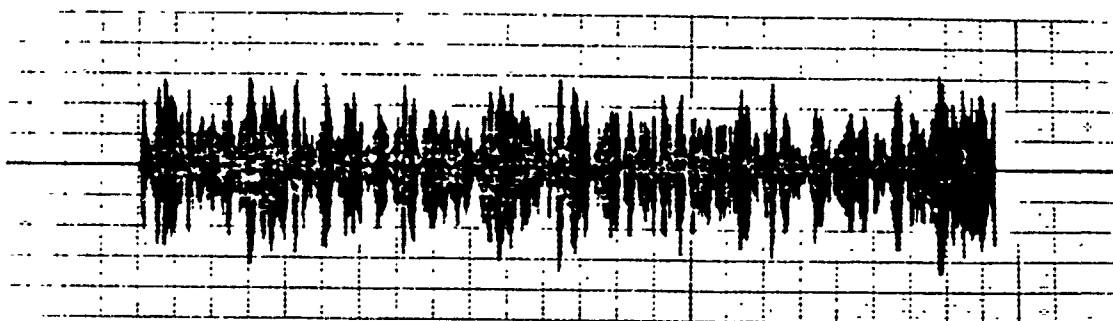


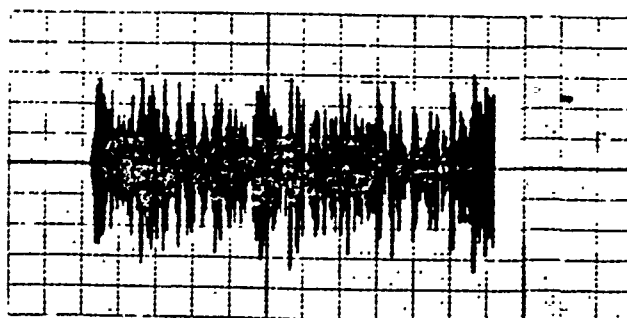
Figure 4-3. Vertical Blade Frequency Displacement of a Ship's Stern Recorded at 25 mm/sec.



5 mm/sec



$2 \frac{1}{12}$ mm/sec (125 mm/min.)



1 mm/sec

Figure 4-4. Vertical Blade Frequency Displacement of a Ship's Stern Recorded at Various Slow Speeds

Before proceeding further, it will be helpful to define three kinds of "peaks" associated with most ship vibration records, and clarify the terminology:

1. In each cycle, the "peak" value is the difference between the mean and the greatest value in that cycle. In this report "peak" will be used in this manner.
2. A modulating signal "peaks" every few cycles. To avoid confusion this report will refer to this type of peak as a "maximum" value.
3. The "peak" spectrum which is really the spectrum of the greatest RMS values in each frequency interval found among several segments. Such values will be called "peak RMS" values.

If the "average RMS" and "peak RMS" values are multiplied by 2 we will obtain spectral single amplitudes which we will call the "average spectral" and "peak spectral" values. Sample spectra are given for three different measurements on a ship in Figure 4-5.

There will be some variation in results, depending on the frequency range used. A higher frequency range will involve broader frequency intervals and yield higher results. The differences will be most apparent when the speed of the shaft or machine being measured varies slightly.

To eliminate the ill effects of speed variations, some analyzers will track a signal and display the spectrum as harmonic components of the tracked signal.

It is obvious that the average spectral value will always be less than the MRV, and that the peak spectral values will be closer to the MRV. The exact relationship between the latter two will depend on the rate of modulation compared to the length of the segment. If the modulation is very slow the amplitude will be near its maximum for the entire length of some segments, and the two will be close. If the modulation is fast, the peak will be closer to the average amplitude.

4.3.4 Histograms of Instantaneous Values

Some analyzers, such as the Nicolet Model 660A Dual Channel Analyzer, sample the signal and obtain histograms of the instantaneous values. This capability might be useful with ship vibration records, but there are several considerations.

First, we are usually concerned with obtaining the amplitudes of the blade or 2 x blade component by itself, so that filtering is necessary as it was for the envelope method. The limitations and corrections discussed in that section apply here also.

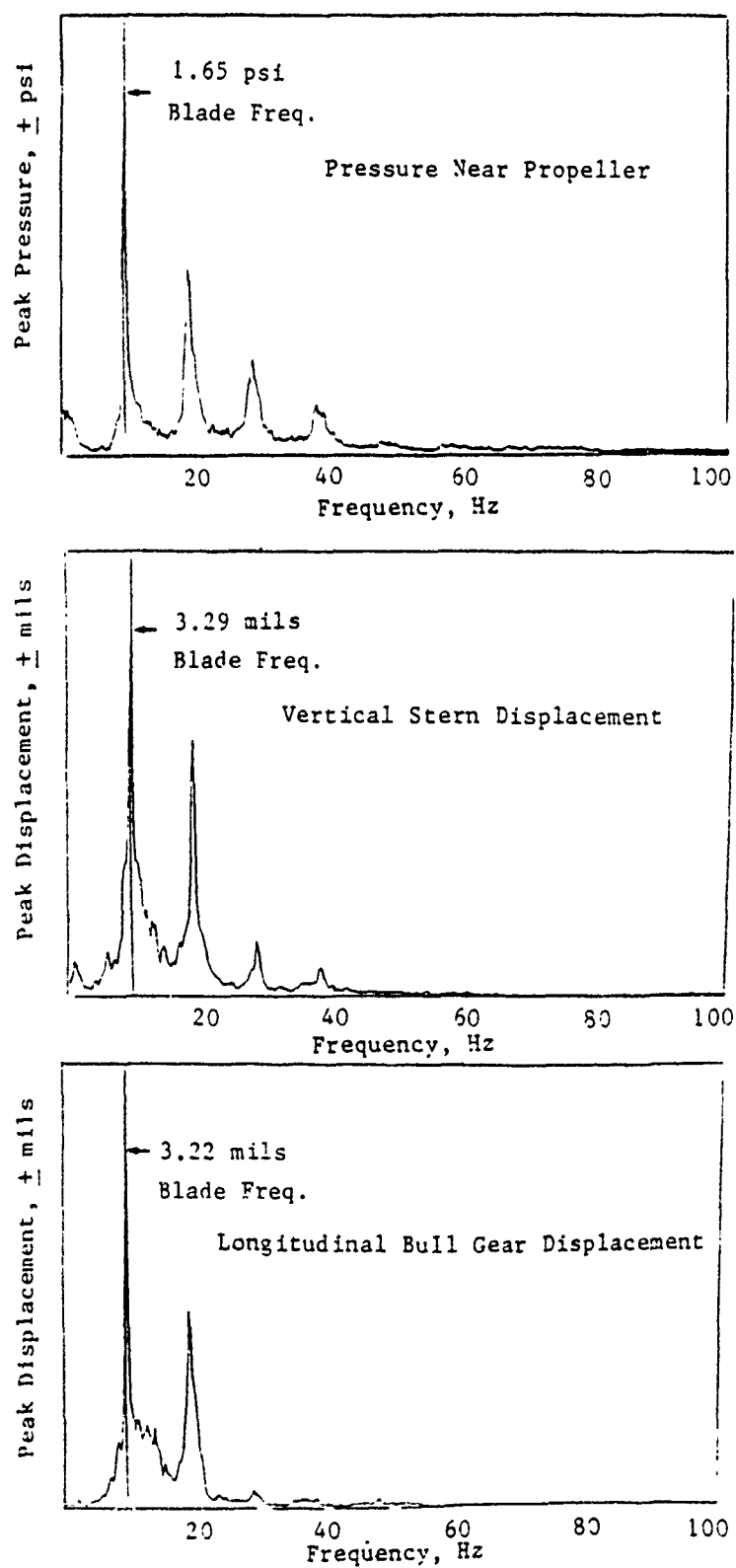


Figure 4-5. Sample Peak Spectra for Various Locations

Second, we are concerned with the peak amplitudes of the cycles. The histograms are usually obtained by sampling all the points on the record, not just the peaks. The amplitude which is exceeded by only 1 or 2 percent of the samples would probably involve only the tips of the largest cycles, and may be comparable to the MRV. The amplitude which is exceeded by some percentage of the samples could be determined from a cumulative probability plot, such as shown in Figure 4-6.

4.3.5 Histograms of Peak Values

This procedure would be closer to the envelope method and can be accomplished with an analog to digital (A/D) converter and a microcomputer in conjunction with filters. It entails sampling the filtered signal, as described for histograms of instantaneous values, then finding the peak (and trough) of each cycle. The peak amplitudes are then put into a histogram or cumulative probability plot. The top 3, 5, or 10 percent of the peak amplitudes may be comparable to the MRV.

The authors are not aware of this procedure being used for routine analysis of ship vibration data, but it would seem to be the most accurate and efficient method of those discussed.

4.3.6 Reporting Formats

The format chosen for reporting data will depend on the purpose of the trials and the type of analysis chosen. For propeller excited vibration the most useful data are usually plots of vibration amplitudes, whether they be maximum repeatable values, average or peak spectral values, or RMS amplitudes, versus RPM. They reflect resonances encountered and any ranges of high vibration levels due to things such as cavitation. When frequency spectra are used, normally there are too many to include all of them in a report, but a few well-chosen examples can help in understanding the nature of the data. Data in tabular form is appropriate for the amplitudes measured during maneuvers. Plots of mode shapes, if any were determined, should be included.

Often measured data is not sufficient to establish accurate mode shapes, but they may be known from vibration analyses. This is particularly true in the case of machinery torsional mode shapes. Such analyses should be utilized in extrapolating data where possible, with sound judgment exercised regarding the validity of such extrapolations. Data generated in this manner should be annotated to reflect how it was obtained.

ISO/DIS 4867 (Reference 4-2) recommends that reported data include the following:

- a. The principal ship design characteristics.
- b. Sketch of Inboard profile of hull and superstructure.
- c. Lines plan of the stern configuration for about one-fifth of the length of the ship.

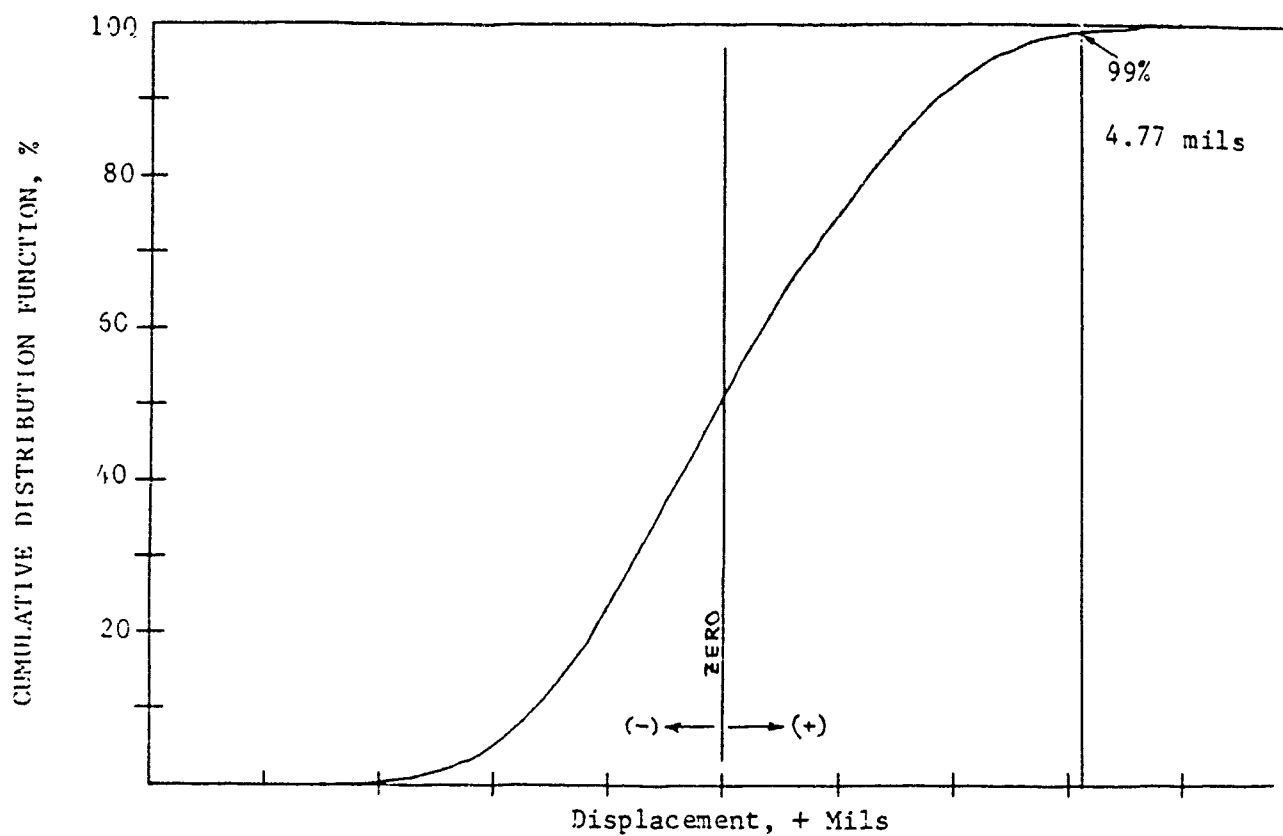


Figure 4-6. Typical Cumulative Distribution Plot with Cursor Set for 99 Percent Probability

- d. Sketch showing locations of hull and machinery transducers. Transducer locations for local vibration measurements should be shown on a separate sketch.
- e. Trial conditions.
- f. Curves of displacement, velocity, or acceleration amplitude versus shaft speed for shaft rotational frequency or blade rate (or machinery excitation frequency) or any harmonic thereof.
- g. Results of measurements at local areas.
- h. Results from maneuvers tabulated.
- i. Results of an anchor drop test including the identified hull natural frequencies, and, from the decaying vibration traces, the derived damping coefficients. Presentation of oscillograph traces is desirable.
- j. Method of analysis of the results.
- k. Type of instruments used.
- l. Hull natural frequencies and modes which have been identified. Also any undesirable or unusual vibration condition encountered.

4.4 TRANSDUCER LOCATIONS

The locations of transducers chosen will depend on the type of trial being conducted, the type of ship being tested, and how thorough a test is desirable. A minimal set of locations for routine sea trials would include:

- Hull Stern
- Thrust Bearing

If there is any suspicion of longitudinal shaft vibration problems, or just to be prudent, the following should be added:

- Main Propulsion System, Longitudinal

When the ship has a large deckhouse aft, and when it is located above large holds or machinery spaces with a minimum of transverse or longitudinal bulkheads to support it, the following should be added:

- Deckhouse

If hull girder vibration is a potential problem or if the hull girder excites a local structure the hull modes should be identified:

- Hull Girder

In addition, a problem not associated with the above measurements may have been identified and diagnostic data may be desired. Some comments on the most common problems are included:

- Rotating Machinery
- Resonant Equipment
- Main Propulsion System, Torsional
- Main Propulsion System, Lateral
- Local Structures

Each of the above is discussed in some detail in this section. Much of the material, particularly for the more routine measurements, is taken from ISO/DIS 4867 (Reference 4-2). The standard locations given can often be directly compared to various criteria presented in Section 8.

4.4.1 Hull Stern

Vertical, athwartship, and longitudinal measurements of the hull girder should be made as close as possible to the centerline and the stern. The steering gear foundation is a recommended location. These measurements should be used for reference purposes. When a torsional response of the hull is to be determined, a pair of deck-edge transducers for vertical vibration should also be employed. It should be ensured that the vibration of the hull girder is measured, excluding local effects.

Normally, velocity gages, integrated to yield displacement, are appropriate for this data. If the lower hull modes are particularly important, accelerometers with low frequency response should be used, perhaps in addition to the velocity gages.

4.4.2 Thrust Bearing

Measurements in three directions (vertical, athwartships, and longitudinal) should be made on top of thrust bearing housing. Recording should also be taken on one supplementary point on the thrust block foundation, in the longitudinal direction.

Blade and twice blade frequency are of primary concern, making the velocity gage the best transducer for the thrust bearing. Data is usually given in terms of displacement.

4.4.3 Main Propulsion System, Longitudinal

To determine the response of the propulsion shaft system to propeller excitation for steam propulsion plants having a reduction gear system, longitudinal measurements should be made at the following locations:

1. Thrust Bearing Housing. The thrust bearing may be located forward or aft of the reduction gear on the same foundation, or aft on a separate foundation.
2. Thrust Bearing Foundation.
3. Forward End of Bull Gear Shaft. This location can normally be accessed by a probe spring loaded to ride on the shaft center. The transducer is attached to the probe.
4. Gear Case Foundation. On top of the gear case foundation under the shaft centerline.
5. Gear Case Top. Over shaft centerline.
6. High Pressure Turbine. Attached to HP turbine casing at forward or aft end.
7. Low Pressure Turbine. Attached to LP turbine casing at forward or after end.
8. Condenser. Mounted as low as practicable and as near the fore and aft centerline as possible.

For diesel propulsion plants longitudinal measurements should be made as follows:

1. Thrust Bearing Housing. The thrust bearing may be incorporated into the structure of the engine at the aft end or mounted separately.
2. Thrust Bearing Housing Foundation.
3. Main Engine. Top, forward end.
4. Forward End of Engine Crankshaft.

For gas turbines, measurements should be made at the thrust bearing and its foundation, the gear case top and foundation, and at the forward end of the bull gear similar to the first five items for steam turbine plants.

Again, blade and twice blade frequency are of primary concern, and velocity gages, integrated to give displacement are recommended.

4.4.4 Deckhouse

As a minimum, the locations given in ISO/DIS 4867 (Reference 4-2) should be measured, i.e., vertical, athwartship, and longitudinal measurements at the following locations to determine the overall vibration of the superstructure:

1. Wheelhouse, centerline at front of bridge.
2. Main deck, centerline at front of deckhouse.

When torsional vibration is to be determined, include a pair of transducers to measure torsional motions of an aft deckhouse.

Normally deckhouse vibration occurs in the frequency range appropriate for velocity gages.

4.4.5 Hull Girder

Where required to identify lower hull modes, vertical or athwartship amplitudes should be measured on the main deck or strength deck level, as close to the centerline as possible, at a sufficient number of points to permit determining the approximate mode shapes of all measured frequencies. If torsional modes are to be defined, phased deck-edge measurements are required. In all cases, structural "hard spots" should be selected. If instrumentation permits, both a roving pickup and a fixed pickup at the stern should be used to simplify the location of nodes by detecting phase changes and providing relative amplitude data. Even better, if enough transducers are available, all points can be measured simultaneously.

Velocity gages can be used except for the lower hull modes, where accelerometers with a low frequency response may be required.

4.4.6 Rotating Machinery

In this section we will consider rotating machinery mounted by means of a foundation, which may or may not have resilient mounts, and attached to a deck, bulkhead, or the hull itself. Excessive vibration may be caused by self-excitation, such as in a rotating machine, or it may be caused by the deck (or whatever the foundation is attached to) vibrating, and the machine or equipment being near resonance on that mount.

In order to provide enough information for proper diagnostics, vertical, horizontal (lateral), and longitudinal measurements should be made on the bearing caps of rotating machinery. If the rotor is relatively long, or if the machine consists of two rotors, such as a turbine and a generator, connected by a coupling, measurements should be made at both ends.

Alternatively, if the foundation and casing are known to be very rigid, and the motion of the shaft only is of interest, vertical and horizontal (later

a) measurements could be made by proximity probes if the shaft is accessible and if a suitable mounting can be devised.

For most rotating machinery problems, velocity gages are preferred. Many of the criteria for acceptability are given in terms of velocity levels. For high frequency response, such as in bearing diagnostics, acceleration is better.

4.4.7 Resonant Equipment

If a piece of equipment is resonating on its foundation, it may be oscillating in any or all of the three translation directions and any or all of the three rotational directions. To detect motion in all directions it would be necessary to have six transducers, where pairs, oriented in the same directions, might be used for rotations. To determine all the motions of the deck (or whatever the machine is mounted to) would take another six transducers. Usually, enough is known about the problem, such as the direction of excessive vibration, that some or even most of these can be eliminated. In any case, the resonant condition, whether it be excited by the machine or equipment itself, or by motion of the base, can be easily detected by the relative motion between the equipment and its base.

The frequencies encountered in resonant situations are normally in the range of velocity gages. Often displacement signals are best for these cases.

4.4.8 Torsional Vibration

If torsional measurements are to be made, it should be done with a thorough knowledge of the expected natural frequencies and mode shapes. This is true because of two factors. First, the number of locations to make torsional measurements is usually very limited, and the most has to be made of what is available. Secondly, the mass and stiffness characteristics in the torsional direction are usually amenable to accurate determination, making the prediction of natural frequencies and mode shapes reasonably reliable. Hence, a measured quantity at one location can be extrapolated by means of these predictions to obtain displacements or stresses at other locations. While this procedure can get complicated and normally requires a vibration engineer, there are several general points that can be provided for guidance.

The two types of measurements generally available are torsional motion (displacement, velocity or acceleration) and torsional strain. If motion is to be measured by means of a torsionmeter it must be mounted on the end of a shaft on its centerline or mounted on an auxiliary shaft driven by a belt off the main shaft. An alternative is to mount an accelerometer or velocity gage to the shaft in a tangential direction.

Torsional displacements are preferred to velocities or accelerations because they can more readily be related to stress.

Torsional strain can be measured by means of strain gages mounted to the shaft.

Whatever locations are chosen they should have relatively high amplitudes as indicated by the calculated mode shapes for the modes that fall within the frequency range of interest. This will minimize errors in the extrapolation process.

4.4.9 Main Propulsion System, Lateral

Lateral measurements are determined by the type of problems encountered and the type of equipment involved. It is difficult to generalize as far as transducer locations are concerned. The following locations are recommended in ISO/DIS 4867 (Reference 4-2).

For lateral vibration of the shaft, vertical and athwartships vibration measurements should be made at least on the shaft relative to the stern tube. It may also be taken relative to line shaft bearings. In order to eliminate possible error, shaft run-out should be checked by rotating the shaft with the turning gear and recording the first-order signal. This signal should be phased and the shaft vibration measurement corrected accordingly.

For lateral vibration of turbines and gears, see Section 4.4.6, Rotating Machinery.

For lateral vibration of diesel engines, vertical and athwartships measurements on the top, forward and aft ends, of the main engine are required as a minimum. Vertical and athwartships measurements are also recommended on the forward and aft ends of the engine foundations.

4.4.10 Local Structures

The discussion for resonant equipment is applicable to this section as well. The only thing that can be readily added is a reminder that if the local structure that seems to be a problem is not a rigid body supported by a foundation, but is instead a flexible member in which a part is vibrating excessively, then the natural frequencies and mode shapes of that structure must be studied, requiring enough transducers to define its mode shapes.

4.5 TEST CONDITIONS

In order to obtain ship vibration data which can be evaluated against existing standards, measurements should be made during uniform test conditions. The discussion of the factors affecting vibration levels and the recommended test conditions are excerpted from ISO/DIS 6867 (Reference 4-2).

The relatively uniform vibration resulting from propulsion machinery excitation (turbine or diesel drive) can be masked or distorted by transient vibrations due to wave impact or slamming. Changes in wake distribution due to rudder angle and yaw can produce large increases in exciting forces. Operation in shallow water also has a significant effect on hull vibration. Propeller emergence, whether continuous or periodic, causes large increases in exciting forces.

In view of the above, the following test conditions are recommended:

1. The test should be conducted in a depth of water not less than five times the draft of the ship.
2. The test should be conducted in a quiet sea, generally State 3 or less.
3. The ship should be ballasted to a displacement as close as possible to the operating conditions. The draft aft should insure full immersion of the propeller.
4. During the free-route portion of the test, the rudder angle should be restricted to about 2 degrees port or starboard (minimum rudder action is desired).

4.6 TEST PROCEDURES

For any test, the first step is to calibrate the recording equipment. If ship control is involved communications must be set up. The procedures for taking data are discussed for tests involving ship control (Hull and Main Propulsion System) and those involving auxiliary machinery. Other tests may require different procedures.

4.6.1 Calibration Procedures

Calibration procedures are categorized as system calibration or electrical calibration. In general, system calibration refers to a procedure which is done before installing instrumentation on board ship, or as the transducers are installed. It should be a complete reckoning of the sensitivity of the transducers, signal conditioning, and recording equipment.

Electrical calibration refers to a procedure which can be accomplished usually at the recording center, is considered a "spot check", takes only a few minutes, and can be done periodically during the vibration trials.

Calibration procedures are different for different types of gages, and are discussed in this section.

4.6.1.1 Accelerometers

All accelerometers can be calibrated over the frequency range of interest by mounting on a shake table or calibration device that is oscillating at known amplitudes. Normally this is the type of system calibration that is used. Strain gage and piezoresistive accelerometers can be calibrated for zero, +1, and -1 g's by laying them on their sides, their bases, and upside down respectively. This provides a D.C. calibration only and is useful only if the conditioning and recording equipment operates at a frequency of zero Hz.

Once the transducers are installed, the "electrical calibration" is usually accomplished by an internal (to the amplifier) signal of known value

being applied to the conditioning and recording equipment. In the case of strain gage and piezoresistive accelerometers, a shunt resistor can be applied across one arm of the bridge, and the value of the resistor can be equated to a certain acceleration. The latter results in a D.C. step being recorded.

4.6.1.2 Velocity Gages

System calibration for velocity gages should be done on a shake table that oscillates at known amplitudes and frequencies. There are no D.C. types of calibration suitable for these gages.

Electrical calibration is done by means of an internal signal of known value being fed into the conditioning and recording equipment. Since the conditioning equipment usually does not operate for D.C. signals, the known signal is normally a sinusoid.

4.6.1.3 Proximity Probes

Proximity probes should not be sensitive to changes in frequency, and therefore can be calibrated for D.C. steps only. They should be calibrated at several distances from the target by means of "feeler" gages. Plastic feeler gages are available that do not affect the signal and can be left in place while recording. Preliminary calibration can be done before installing the probes on board ship, but the final calibration should be done with the probes in place because each target has a slightly different effect on the gage.

The only practical type of "electrical" calibration would be the substitution of a signal of known value.

4.6.1.4 Strain Gages

The type of calibration used with strain gages will depend on how they are used. Again the signal output should not be sensitive to frequency, and D.C. calibrations are adequate. If possible, the strain gaged object should be subjected to known loads, and the resulting strains calculated and related to the signal output. If the object is a complex shape, or if known loads are difficult to apply, the only choice is to accept the manufacturers listed Gage Factor and use a shunt resistor for calibration.

In most applications on board ship, the signal leads from the strain gages are fairly long. This reduces the sensitivity of the gages, requiring the shunt to be applied at the gage rather than at the amplifier to obtain accurate results.

4.6.2 Communications

The trial director, who should be stationed at the recording center, should have communications by sound powered phones or other means with the bridge or the engine control center, whoever is controlling the course and speed of the ship. Whoever is on the phones at the controlling station should have access to rpm gages and a rudder indicator and be able to advise the trial

director immediately of any changed conditions. Often the trial director will station himself and his equipment in a space where that information is available directly.

4.6.3 Hull and Main Propulsion System Vibration

ISO/DIS 4867 (Reference 4-2) and SNAME Code C-1 (Reference 4-3) give test procedures for gathering data on hull and main propulsion system vibration in commercial ships:

- a. Make a steady deceleration or acceleration run of, preferably, less than 5 rpm per minute to determine location of critical speeds.

NOTE: These runs do give an indication of critical speeds, but if the change in shaft speed, which is hard to control, is uneven, when the propeller is loaded it may give a false indication of resonance. Also, the amplitudes cannot be trusted since steady-state conditions have not been established, and only a small sample is considered at each speed.

- b. In free route, run from half shaft speed to maximum speed at increments of 3 to 10 rpm. Additional runs at smaller increments are required in the vicinity of critical speeds and near-service speed.
- c. Hard turns to port and starboard at maximum speed (optional).
- d. Crashback from full power ahead to full power astern (optional).
- e. Anchor drop-and-snub (optional).

For steady speed free-route runs, permit ship to steady on speed. Hold at steady speed for a sufficient time to permit recording of maximum and minimum values (about one minute). In multiple shaft ships, all shafts should be run at, or as close as possible to the same speed to determine total vibration amplitudes. In certain instances it may be preferable to run with a single shaft when determining vibration modes.

NOTE: A one minute record length was recommended with oscillographic analysis in mind. For electronic analysis, which is less time consuming, 2 to 3 minutes is recommended.

For maneuvers, start the recorder as the throttle or wheel is moved. Allow to run until maximum vibration has passed. This normally occurs when the

ship is dead in the water during a crashback maneuver or when the ship is fully in a turn.

For the anchor drop-and-snub test, the anchor must fall freely and be snubbed quickly by use of the windlass brake, and must not touch bottom. The ship must be dead in the water for this test, with a minimum of rotating equipment in operation. Care must be taken not to exceed the recommendations for free drop as indicated by the manufacturer of the anchor windlass. Data should be taken continuously from the moment the anchor is released until vibration can no longer be detected.

4.6.4 Auxiliary Machinery

ISO/DIS 4868 (Reference 4.4) and SNAME Code C-4 (Reference 4.5) give test procedures for gathering data on local shipboard structures and machinery. When evaluating the vibration of auxiliary machinery, the following guidelines are recommended:

- For constant speed units, measurements should be made at the rated speed.
- For variable speed units, measurements should be made at about five equally spaced points in the operating speed range, including known criticals.
- For multi-speed units, measurements should be made at each operating speed.
- To minimize interference, as much nearby equipment as possible should be shut down.

NOTE: To familiarize personnel with the measurement of shipboard vibration, the simple mechanical instruments referred to in Section 4.1.1 are recommended. As a next step, instrument packages are available which include a transducer, tunable filter, and calibrated meter, as discussed in Section 4.1.4.1. For more complex studies, including ship trials, the instrumentation system described in SNAME Code C-1 is recommended. This version of shipboard vibration instrumentation is maintained by the Maritime Administration and, under special conditions, may be borrowed from MARAD.

4.7 REFERENCES

- 4-1. Manley, R.G., "Waveform Analysis," John Wiley and Sons, Inc. (1946).
- 4-2. Draft International Standard ISO/DIS 4867, "Code for the Measurement and Reporting of Shipboard Data," 3 Nov. 1982.
- 4-3. SNAME T&R Code C-1, "Code for Shipboard Vibration Measurement," 1975.

- 4-4. Draft International Standard ISO/DIS 4868, "Code for the Measurement and Reporting of Shipboard Local Vibration Data," 1983.
- 4-5. SNAME T&R Code C-4, Shipboard Local Structure and Machinery Vibration Measurements, 1976.

SECTION 5.0

CRITERIA FOR ACCEPTABLE LEVELS OF VIBRATION

In developing useful criteria to be used in the evaluation of shipboard vibration a review will be given of the important points made in Section 1.0, Introduction, and in Section 3.0, Introduction to Ship Vibration. Also, for purposes in this guide one should keep the following points in mind:

- The ship is excited by both transient and periodic forces.
- In most cases transient forces are caused by rough seas.
- Most periodic forces are generated by the ship's propeller and machinery systems.
- Heavy transient forces, such as slamming, will excite structural resonances and can cause serious damage in heavy seas.
- Comparatively low periodic forces, when combined with resonant conditions, can cause serious shipboard vibration problems.
- Both transient and periodic forces are aggravated by heavy seas and hard maneuvers.
- This guide is directed toward the control and attenuation of vibration excited by periodic forces and do not relate to transient excitation.

To properly evaluate shipboard vibration it has been generally accepted that uniform test conditions should be employed for vibration trials, such as those specified in the SNAME T&R Code C-1, "Code for Shipboard Vibration Measurement," and ISO 4867, "Code for the Measurement and Reporting of Shipboard Vibration Data," both of which have similar requirements. Thus, in the absence of other design requirements, a standard method of testing can be employed for all ships, for evaluation against uniform criteria. It should be noted, however, that while these trial conditions will more closely represent design conditions, more serious vibration can be expected under adverse operating conditions and suitable factors must be included in the design of structural and mechanical components to account for the maximum anticipated dynamic stresses.

Shipboard vibration is considered excessive when it results in structural damage, in damage or malfunction of vital shipboard equipment, or adversely affects the comfort or efficiency of the crew. Normally crew complaints will occur before vibration becomes damaging to the ship's structure.

However, failure or malfunction of vital shipboard equipment may occur without significant annoyance to the crew.

The criteria recommended in this guide are based on existing requirements, as they relate to:

- Human reaction (habitability)
- Machinery and equipment malfunction
- Fatigue failure

For convenience the total ship system relates to the five basic elements defined in Section 5.0 in the following manner:

5.1 General Hull Vibration

Most shipboard vibration problems originate with the vibration of the hull (ship's girder). The criteria recommended relates to human reaction.

5.2 Major Substructures and Local Structures

These structures, which are attached to and excited by the hull girder, can relate to all three criteria.

5.3 Machinery Vibration

In most instances, machinery vibration relates to malfunction or fatigue failure of components.

5.1 GENERAL HULL VIBRATION

The recommended criteria for general hull vibration is based on human reaction to the vibration environment normally experienced aboard ship, and may be used to evaluate hull and superstructure vibration in normally occupied spaces. This criteria, shown in Figure 5-1, is based on maximum repetitive values (peak values) for each component such as shaft frequency, propeller blade frequency, or harmonics of propeller blade frequency, and is identical to those in ISO 6954 and SNAME T&R Bulletin 2-25. The curves are shown in both metric and English units.

For convenience of interpretation, Figure 5-2 shows a linear displacement plot of a 4 mm/sec or 0.16 in/sec constant velocity curves, which represents the lower limit of the shaded area of Figure 5-1, above 5 Hz. The 9 mm/sec or 0.36 in/sec velocity curve represents the upper limit of the shaded area of Figure 5-1, above 5 Hz. Below the 4 mm/sec curve, referred to as Zone I by the SNAME guidelines, adverse comments are generally unexpected. Above the 9 mm/sec curve, Zone III, complaints are generally expected. Zone II, which represents the shaded area in the guideline curves, has been further divided by

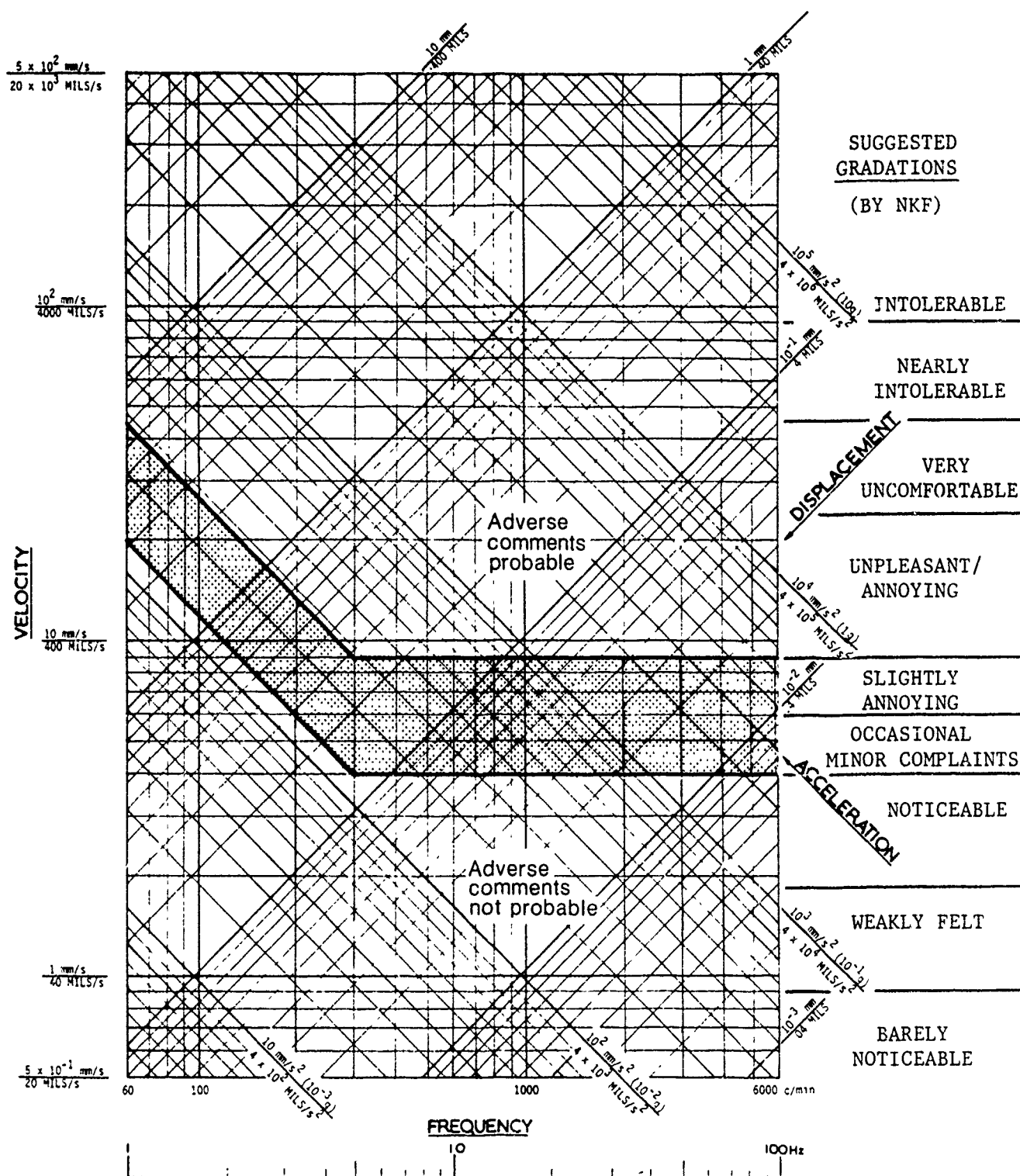


Figure 5-1. Guidelines for the Evaluation of Vertical and Horizontal Vibration in Merchant Ships (Peak Values)*

* Peak value is the maximum repetitive value, see ISO 4867

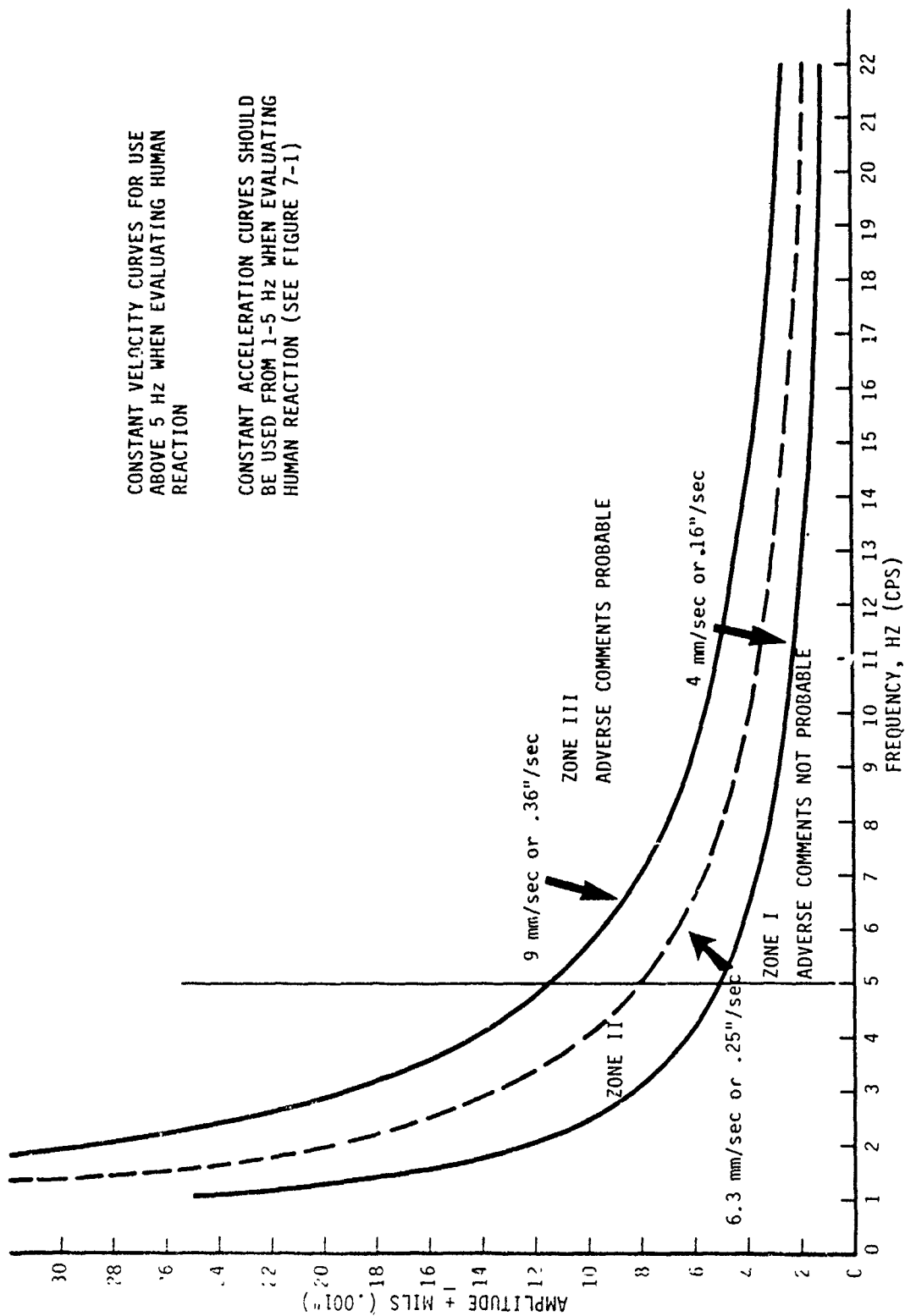


Figure 5-2. Guidelines for Ship Vibration - Vertical and Horizontal

a 0.25 in/sec or 6.3 mm/sec curve to represent a finer evaluation of complaints received. It is recommended that vibration levels in Zone I be considered totally acceptable from 5 to 100 Hz. Vibration levels in Zone III generally require further investigation. Vibration levels in the upper half of Zone II (above 0.25 in/sec or 6.3 mm/sec) may require further investigation if personnel are exposed to these levels for extended period of time (above 8 hours). Below this curve, complaints would be considered of minor importance.

Below 5 Hz, the ISO and SNAME guidelines for human reaction show corresponding constant acceleration curves of .013 g for the lower limit and .029 g for the upper limit. While the corresponding amplitudes below 5 Hz would be relatively high (greater than shown on the constant velocity curves of Figure 5-2) the normal excitation at that frequency would result from dynamic or hydrodynamic unbalance in the propulsion system with attendant hull resonances at certain operating speeds (rpm). In great lakes ships, which are long and slender, the fundamental frequency may be below 1 Hz and this may be excited by wave energy that includes that frequency, which could produce springing or resonant vibration at the hull natural frequency. The vibration level would be high but the acceptable limit would be based on the total allowable hull bending stress. Also see Section 5.2.3.1, Hull Girder Vibration (Springing).

For seagoing ships the upper limit of .029 g is recommended below 5 Hz for "corrective action required." Dynamic balancing, or correction of mechanical errors associated with shaft frequency vibration should be investigated. Propeller damage, an out of pitch or bent propeller blade, propeller unbalance, or misalignment are frequent offenders.

As noted in Section 6.0, shipboard vibration generally approximates to narrowband random vibration and a crest factor of 2.5 is commonly encountered during trial conditions. In these circumstances the maximum repetitive vibration is more appropriate than rms vibration with regard to evaluation of overall ship vibration. Both the SNAME guidelines and ISO 6954 evaluate overall shipboard vibration in terms of maximum repetitive values and, for comparison with rms values, the crest factor must be taken into account.

In ISO 2631, the effect of vibration on human beings is evaluated by reference to curves of rms acceleration, taking the evaluation to apply over a wide range of crest factors. Compliance with the guidelines recommended herein, which corresponds to ISO 6954, will also comply with ISO 2631 with respect to crew exposure to whole body vibration as long as the upper band specified, when converted to rms acceleration factors of 1.6 and 3.0, is below the criteria curves selected on the basis of ISO 2631. In this regard, the adequacy of the recommendations given in this guide is shown on Figure 5-3.

5.2 MAJOR SUBSTRUCTURES AND LOCAL STRUCTURES

Based on the general philosophy expressed in Section 5.0, we can divide the complete ship system into a number of its basic elements for convenience in the diagnosis, evaluation, and development of corrective actions in the resolution of shipboard vibration problems. In a similar manner, we can develop the vibration criteria applicable to the specific elements or problem areas. In

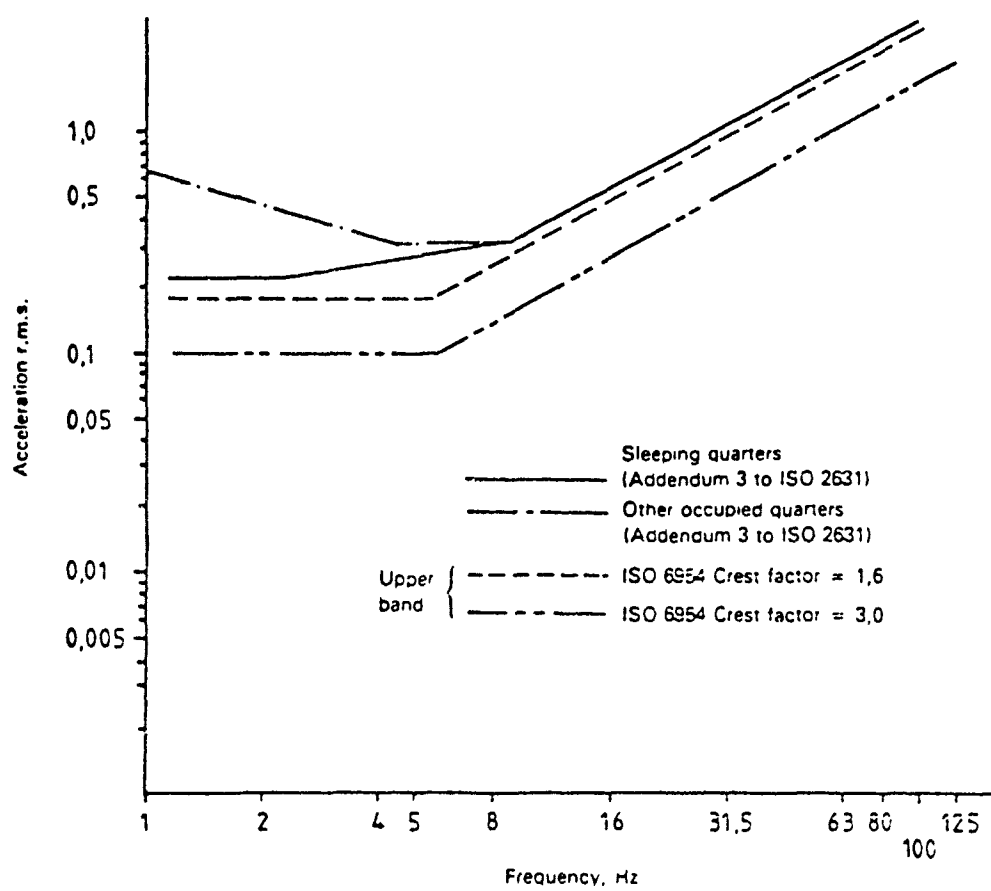


Figure 5-3. Comparison Between ISO 6954 and Addendum 3 to ISO 2631

this regard we have chosen to combine major substructures and local structures together since they both represent segments of the total ship structure in which the hull girder represents the vibrating platform on which these components are attached, frequently in a descending order of structural rigidity. In this area we may encounter all three of the basic criteria; human reaction, equipment malfunction, and fatigue failure.

5.2.1 Human Reaction

The criteria for human reaction throughout the ship remains the same for all areas designated as accommodations or working spaces. Major substructures, such as deckhouses or large deck areas, may magnify the basic hull vibration, or local structures such as a compartment deck in the deckhouse may further amplify the hull vibration. However, we would still use the same criteria for adequacy, based on human reaction. Thus, all areas utilized for habitability purposes should meet the requirements recommended in Section 5.1 for general hull vibration.

5.2.2 Equipment Malfunction

Equipment malfunction or damage may occur as a result of the vibration of those structural components to which the equipment is attached or may be due to the sensitivity of the equipment. Examples may include meters mounted on bulkheads, electronic equipment mounted on isolation mountings, binacles mounted on the bridge deck, equipment mounted on a fabricated foundation, switchboard equipment, transformers, steam piping, etc. In short, when considering the response of passive (non self-exciting) equipment which could result in malfunction or damage to the equipment installed in the ship we have to consider the structural adequacy of the support system and the adequacy of the equipment to perform its function in the shipboard vibration environment.

5.2.2.1 Structural Adequacy of Support System

The structural adequacy of the total support system for any shipboard mounted equipment must be related to the basic hull vibration and the capability of the equipment to adequately perform in a shipboard vibration environment. In Section 5.1 we identified criteria for the evaluation of hull vibration in which we noted vibration levels in Zone III, "Adverse Comments Probable," and also noted that further investigation would generally be required if these guidelines were exceeded.

As a rule of thumb, it is recommended that the structural adequacy of the support system be based on the response of the local structure, at the mounting point, when the structure is loaded as it would be in service and that the vibration amplitude not exceed that of the basic hull structure in that area by more than 50 percent. This limitation would prohibit structural resonance but would allow for some amplification of the local structure with reference to the input motion of the vibrating platform, the hull girder, and would restrict the motion to a maximum of 1.5×9 or 13.5 mm/sec in the frequency range of 5 to 100 Hz, when the maximum recommended limit of 9 mm/sec occurs in the hull.

Frequently, excessive vibration of equipment may be directly related to the geometry of the structural support system and/or the improper use of resilient mountings, thus producing a resonant response. Examples would be a ship's binnacle located on an improperly supported deck section or a tall electronic chassis with resilient mountings placed too close together. In such cases excessive vibration may result, although the observed amplitude at the structural base appears satisfactory. Appropriate corrective action could include modifications to the support system and/or the addition of supporting braces. Similar problems can occur within shipboard equipment, frequently resulting in damage or malfunction in service. Hence, it is considered necessary to ascertain whether the problem is one of resonant structure, faulty installation, or unsatisfactory equipment.

5.2.2.2 Vibration of Shipboard Equipment

Failure or malfunction of shipboard equipment subjected to shipboard vibration is not necessarily caused by excessive vibration at the point of support, as noted above. It has been well established that commercially available equipment, originally designed for stationary installations, frequently fail when used in the shipboard vibration environment. The resonance of components of the equipment must be avoided and the equipment should be qualified in vibration resistance for shipboard use.

To ensure consistency in vibration resistance requirements for shipboard equipment and machinery, the International Organization for Standardization (ISO/TC108/SC2/WG2 Vibration of Ships) has undertaken the development of a Code for Testing and Evaluation of Vibration Resistance of Lightweight Shipboard Equipment and Machinery Components as Work Item No. 82. The draft working paper of November 1981 is based, in part, on MIL-STD-167-1 (SHIPS), Mechanical Vibration of Shipboard Equipment, Type 1, Environmental, and is consistent with the basic environmental testing procedures outlined in IEC Publication 68-2-6, Fifth Edition, 1982, which has, as its objective, "to provide a standard procedure to determine the ability of components, equipment, and other articles to withstand specified severities of sinusoidal vibration."

When investigating equipment failure or malfunction it is necessary to determine:

1. If the vibration of the supporting structure is excessive
2. If the method of supporting the equipment on the structure is magnifying the input motion unnecessarily (resonance)
3. If the equipment is unsatisfactory for the shipboard vibration environment

To assist in the evaluation of the vibration resistance of lightweight shipboard equipment and machinery components under study, the proposed test procedures and test requirements are provided in Section 5.2.3. It should

be noted, however, that these test requirements represent an accelerated vibration test to simulate the environmental vibration which may be encountered aboard ships under adverse conditions. The vibration levels recorded on a ship during vibration trials are lower than the levels shown in Table 5-1. The amplitudes specified for the environmental test are sufficiently large within the selected frequency range to obtain a reasonably high degree of confidence that equipment will not malfunction under the most severe service conditions.

5.2.3 Environmental Testing of Shipboard Equipment

The test specified herein is intended to locate resonances of the equipment and impose an endurance test at each of these resonances. Equipment that passes this test will have a greater probability of satisfactory performance aboard ships.

5.2.3.1 Vibration Tests

Each of the tests specified herein shall be conducted separately in each of the three principal directions of vibration. All tests in one direction shall be completed before proceeding to tests in another direction. The equipment shall be secured to the vibration table and shall be energized to perform its normal functions. If major damage occurs, the test shall be discontinued and the entire test shall be repeated following repairs and correction of deficiencies, unless otherwise directed by the agency concerned. The manufacturer may, at his option, substitute an entirely new equipment for retest. If this option is taken, it shall be noted in the test report.

5.2.3.2 Exploratory Vibration Test

To determine the presence of resonances in the equipment under test, the equipment shall be secured to the vibration table and vibrated at frequencies from 2 Hz (or lowest attainable frequency) to 15 Hz, at a table vibratory amplitude of ± 1.0 mm. For frequencies from 15 to 100 Hz, the equipment shall be vibrated at an acceleration level of ± 0.9 g. The change in frequency shall be made in discrete frequency intervals of 1 Hz and maintained at each frequency for about 15 seconds. The frequencies and locations at which resonances occur shall be noted.

5.2.3.3 Endurance Test

The equipment shall be vibrated for a period of at least 90 minutes at each of the resonant frequencies chosen by the test engineer at the corresponding amplitudes shown in Table 5-1. If no resonances are observed, this test shall be performed at the upper frequency as specified in Table 5-1 for each category for a period of 2 hours.

5.2.3.4 Variable Frequency Test

In addition to the endurance test, the equipment, according to the various categories, shall be tested in accordance with the vibration levels

shown in Table 5-1 or Figure 5-4 in discrete frequency intervals of 1 Hz. At each integral frequency, the vibration shall be maintained for 5 minutes.

Table 5-1. Vibration Test Requirements for Shipboard Equipment and Machinery

| CATEGORY | FREQUENCY RANGE | DISPLACEMENT OR ACCELERATION VALUE* |
|---|-------------------------------|---|
| 1) Control and Instrumentation Equipment when mounted on Diesel Engines, Air Compressors and other severe environments. | 2 to 25 Hz 25 to 100 Hz | ± 1.6 mm (Disp.) ± 4.0 g (Accel.) |
| 2) Communication and Navigation Equipment, Control and Instrumentation Equipment, and other Equipment and Machinery | 2 to 15.0 Hz 15.0 to 50 Hz | ± 1.0 mm (Disp.) ± 0.9 g (Accel.) |
| 3) Mast-Mounted Equipment | 2 to 15.0 Hz 15 to 50.0 Hz | ± 1.0 mm (Disp.) ± 2.25 g (Accel.) |

*Allowable deviation from these values is ± 10 percent.

5.2.3.5 Exception

Category 2 or 3 equipment intended for installation solely on a particular class of ship need be vibrated only up through the frequency range which includes the second harmonic exciting frequency of the propeller ($2 \times$ maximum shaft rpm \times No. of blades $\div 60$).

5.2.3.6 Endurance Test for Mast-Mounted Equipment

Equipment intended for installation on masts, such as radar antennae and associated equipment shall be designed for a static load of 2.5 g (1.5 g over gravity) in vertical and transverse (athwartship and longitudinal) directions, to compensate for the influence of rough weather. In addition, the equipment shall be vibrated for a total period of at least 90 minutes at the resonant frequencies chosen by the test engineer. If no resonance is observed, this test shall be performed at 50 Hz, unless excepted by 5.2.3.5 above. The vibration levels shall be in accordance with those of Category 3 in Table 5-1.

5.2.4 Structural Fatigue Failure

Fatigue failures have been known to occur in major ship structures such as the hull girder, or serious bow damage which can occur in extreme weather conditions. In most cases, however, such failures are the result of

design deficiencies originating in areas of high stress concentrations combined with high dynamic or shock loads. As pointed out earlier, this guide does not relate to high transient forces but to the periodic forces generated by the operation of the vessel and its machinery under normal operating conditions.

Fatigue failure under normal operating conditions can occur in the ship's structure under normal operating conditions when the exciting forces are combined with resonant structural vibration, high stress concentration factors, and low system damping. Specific examples of such failures can include the hull girder, local structures, and equipment supports.

5.2.4.1 Hull Girder Vibration (Springing)

Hull girder vibration, at the fundamental natural frequency of the hull, also referred to as springing, has been found to be a potential problem area on ore carriers on the Great Lakes. This results from a combination of factors which can produce significant dynamic stresses at the hull natural frequency which, when combined with normal loading stresses, can approach dangerous levels.

Unlike oceangoing ships, which can experience dangerous hull stress levels by a combination of loading, heavy seas, and slamming effects which represent transient forces not covered by this guide, Great Lakes ore carriers are longer and more slender, have a relatively lower midship section modulus, and lower natural frequency in bending. In addition, the wave patterns on the Great Lakes are periodic in nature and the periodicity of encounter between the bow of the ship and the waves can excite a resonance of the fundamental hull frequency. Supplemental dynamic loading may also be introduced by the nonlinear excitation caused by the interaction of two different long wave components. As a result, under certain headings and sea conditions or wave trains, the resulting dynamic hull stresses can be excessive.

Much study has been made on this subject and care must be taken to avoid this resonant phenomena by making necessary adjustments to hull speed and/or direction of encounter with the waves. For purposes of this guide, however, we would recommend adherence to the human reaction or habitability criteria as given in Figure 5-1. Dangerous hull stresses will not occur within an estimated maximum allowable amplitude of ± 25 mm (± 1.00 inch). As an alternative, stress monitoring based on design analyses should be employed.

5.2.4.2 Local Vibration

The majority of structural fatigue failures that occur aboard ship are related to resonant vibration of local structural members which are readily recognizable. Typical cases include supports to radar antennas, equipment supports, handrails, etc. In most cases the problem is recognizable and may be readily corrected by stiffening the support structure so that resonance does not occur below 115 percent of operating speed (rpm).

Not so obvious are fatigue cracks which may develop in the aft peak tank and adjacent structures. Most of such cracks can be related to

propeller pressure forces generated by cavitation effects and resonant local structural elements with high stress concentration factors. The immediate correction would involve stiffening of the resonant member and the correction of stress concentration points. Depending on other problems aboard the ship, consideration might be given to the correction of the exciting forces. If this approach is taken, a maximum hull pressure force of ± 8 kPa or ± 1.16 psi, measured on the centerline, over the propeller, is recommended.

5.3 MACHINERY VIBRATION

Shipboard machinery includes the main propulsion machinery, auxiliary machinery, support machinery, and related equipment. In this category we are primarily concerned with the effects of vibration on system dynamics (fatigue failure of components) and the environmental effects on machines and equipment (damage and/or malfunction). In this latter area, we are concerned with active shipboard equipment and its self-generating forces. The subsections include Main Propulsion Machinery, which relates to fatigue failure of components, and General Machine Vibration, which relates to environmental effects.

5.3.1 Main Propulsion Machinery

The main propulsion machinery includes all components from the engine up to and including the propeller, and thus contribute to the vibration of the ship and to dynamic stresses within the propulsion system itself by forces generated both by the propeller and by the propulsion system components. Recommended criteria to be employed in the control of the more important dynamic forces existing in the main propulsion system are based on design requirements. For further background and explanation of system forces and response, refer to Section 5.6.

It should be noted that vibration measurements alone cannot always be used to determine the acceptability of dynamic systems. The levels of dynamic stresses are dependent on both the vibration amplitude and the dynamic analysis of the vibrating system. Thus, the evaluation of most machinery vibration characteristics will require an experienced investigator capable of performing detailed design analyses and complex vibration measurements.

The main engines, shafts, couplings, reduction gears, propellers, and related equipment are designed for structural adequacy when operating under the conditions stipulated in the procurement specifications. The vibration characteristics of the propulsion system must be controlled to avoid the presence of damaging vibratory stresses within the system, as well as the generation of severe hull vibration. Potential problem areas include unbalance and misalignment of system components; excessive shaft stresses; and longitudinal, torsional, and lateral vibration of the propulsion system.

5.3.1.1 Dynamic Unbalance and Misalignment

All rotating propulsion machinery shall be balanced to minimize vibration, bearing wear, and noise. The types of correction, as shown in Table

5-2 below, shall depend on the speed of rotation and relative dimensions of the rotor.

Table 5-2. Types of Correction

| TYPE OF CORRECTION | SPEED (RPM) | ROTOR CHARACTERISTICS |
|--|---------------------|--|
| Single-Plane | 0 - 1000 0 - 150 | $L/D \leq 0.5$ $L/D > 0.5$ |
| Two-Plane | > 1000 > 150 | $L/D \leq 0.5$ $L/D > 0.5$ |
| Multi-Plane | | Flexible: Unable to correct by two-plane balancing |
| L = Length of rotor mass, exclusive of shaft D = Diameter of rotor mass, exclusive of shaft | | |

The residual unbalance in each plane of correction of any rotating part shall not exceed the value determined by:

$$U = \frac{4W}{N} \quad \text{for speeds in excess of 1,000 rpm}$$

$$U = \frac{4000W}{N^2} \quad \text{for speeds between 150 and 1,000 rpm}$$

$$\text{or } U = 0.177W \quad \text{for speeds below 150 rpm}$$

where: U = Maximum residual unbalance in ounce-inches

W = Weight of rotating part in pounds

N = Maximum operating rpm of unit

When checking the propulsion system for first-order (shaft rpm frequency) forces, in addition to balancing, check propeller for pitch accuracy, shafting and couplings for run-out or bending, and stern bearings for uneven or excessive wear. Shafting should also be checked for corrosion/fatigue cracks originating in keyway fillets.

5.3.1.2 Dynamic Shaft Stresses

Conventional design requirements for propulsion shafting generally includes factors to compensate for the eccentric thrust produced at the propeller. This eccentric thrust produces a dynamic bending moment, due to shaft rotation, with maximum alternating bending stresses usually occurring at the propeller keyway. This dynamic stress is greatly influenced by the actual moment arm between the propeller and the effective point of support of the aftermost bearing, and the presence of seawater which represents a corrosive medium and greatly reduces the fatigue characteristics of the shaft. These stresses are also significantly effected by sea and operating conditions and are the root cause of most shaft failures which have occurred.

If, during normal maintenance procedures, evidence of fatigue cracks in the tailshaft, in the vicinity of the forward face of the propeller, are noted, it would be prudent to check the alternating bending stress of the tailshaft against the following empirical formula:

$$C \frac{(M_g + M_t)}{6000}$$

where: S = Section modulus = $\frac{I}{R}$

C = Service factor = 1.75 for commercial ships

M_g = Gravity moment due to overhanging propeller weight calculated from forward face of propeller hub to assumed point of shaft support (1 diameter of shaft for water lubricated bearing and 1/2 diameter for oil lubricated)

M_t = Calculated moment of eccentric thrust = 0.65 x Propeller Diameter x Rated Thrust

I = Shaft moment of Inertia

R = Shaft radius

6000 = Maximum safe fatigue limit to be used for the assembly operating in the presence of a corrosive medium (psi)

Cold rolling the tailshaft in the vicinity of the keyway forward beyond the aft end of the liner has been found to be effective in retarding the propagation of fatigue cracks. A detailed dynamic analysis of the complete propulsion system is strongly recommended, particularly in the case of diesel drive systems or new or unusual design concepts.

5.3.1.3 Torsional Vibration

The mass-elastic system, consisting of engine, couplings, reduction gears, shafting, and propeller, should have no excessive torsional vibratory stresses below the top operating speed of the system nor excessive vibratory torque across the gears within the operating speed. Excessive torsional vibratory stress is that stress in excess of

$$S_v = \frac{\text{Ultimate Tensile Strength}}{25}$$

Below the normal operating speed range, excessive torsional vibratory stress is that stress in excess of 1-3/4 times S_v .

Excessive vibratory torque, at any operating speed, is that vibratory torque greater than 75 percent of the driving torque at the same speed, or 10 percent of the full load torque, whichever is smaller.

Gear rattling is a strong indication of torsional vibration in a geared drive. To evaluate any torsional vibration measurements, it will be necessary to have available, or to develop, a complete mathematical analysis of the system to be tested. It is obvious that experienced personnel are required to conduct such studies.

5.3.1.4 Longitudinal Vibration

Longitudinal vibration of the main propulsion system is frequently a problem and can cause significant structural vibration within the ship. It may be very pronounced at the main thrust bearing, at other parts of the propulsion system, and particularly in the higher levels of deckhouses. If significant vibration in the fore-and-aft direction is noticed, the problem should be investigated.

To avoid damage or crew annoyance, the propulsion system should have no excessive alternating thrust within the operating speed range. In no case, however, should the displacement amplitude of longitudinal vibration of the propulsion machinery, including the main condenser and associated piping in a steam turbine drive, be sufficient to adversely affect the operation of the propulsion unit or precipitate fatigue failure of components such as thrust bearings or gear teeth. Pitting of gear teeth may also indicate excessive torsional or longitudinal vibration.

Excessive alternating thrust is defined as:

(a) Main and turbine thrust bearings

Excessive alternating thrust occurs when the single amplitude of alternating thrust, measured at the main and turbine thrust bearings, exceeds 75 percent of the mean thrust at that speed or exceeds 25 percent of the full power thrust, whichever is smaller.

(b) Excessive alternating thrust

Excessive alternating thrust In the reduction gear occurs when the vibratory acceleration of the bull gear hub exceeds ± 0.1 g unless another value is provided by the gear manufacturer. If the acceleration exceeds the allowable value, calculations will be required to determine the vibratory stresses in the gear teeth to determine their acceptability to the gear supplier.

(c) Excessive longitudinal vibration

Excessive longitudinal vibration of the main propulsion system components (including condenser, piping, etc.) occurs when the vibration exceeds ± 0.25 g, or that level certified as satisfactory by the equipment manufacturer, whichever is the least.

Although detailed measurements would be required to evaluate the presence of excessive longitudinal vibration in (a) or (b) above, the hammering of the thrust bearing would represent a very dangerous condition and must be avoided until properly investigated. As in the case of excessive torsional vibration, gear rattling may also occur if the longitudinal vibration is excessive. In some instances, particularly in diesel drives, harmonic components of torsional and longitudinal vibration may be coupled through the action of the propeller.

5.3.1.5 Lateral Vibration

Lateral vibration in the main propulsion shafting could be destructive if the fundamental frequency is resonant in the operating speed range. The fundamental frequency, sometimes referred to as "whipping", occurs at shaft rpm and is excited by propeller and shafting unbalance. In all designs this fundamental frequency must occur well above operating speed (115 percent of maximum rpm). Frequency can be effected, however, by misalignment, bearing wear-down, or lost bearing support (structural failure). The presence of large or increased lateral amplitudes would be an apparent symptom and should be investigated.

Whirling frequencies at blade rate frequency are excited by propeller forces at \pm the shaft rate. Thus, a five-bladed propeller would excite fourth and sixth order frequencies, referred to as counterwhirl and forward whirl, respectively. However, these frequencies are not generally significant because of the low level of propeller forces normally encountered. It is usually customary to avoid the presence of the frequencies in the upper 15 percent of the speed range. However, if strong response occurs at these whirling frequencies, further investigation would be recommended.

5.3.2 General Machine Vibration

Shipboard machinery is referred to in this guide as "active" shipboard equipment since, in addition to being affected by general hull vibration, generates vibratory forces which contribute to the total motion of the machine itself and may also adversely effect the structure to which it is attached. The maximum acceptable vibration of shipboard machinery is frequently defined by the manufacturer. When this information is available, it should be used. When such information is not available the criteria provided herein is recommended.

5.3.2.1 Nonreciprocating Machines

The maximum allowable vibration of rotating machinery required to demonstrate compliance with MIL-STD-167-1 (SHIPS) balancing requirements is shown in Figure 5-5. On all machinery except turbines, amplitudes of vibration are measured on the bearing housing in the direction of maximum amplitudes. In the case of turbines, amplitudes of vibration are measured on the rotating shaft adjacent to the bearings. When feasible, machinery is completely assembled and mounted elastically at a natural frequency less than one-quarter of the minimum rotational frequency of the unit. Large and complex units are shop tested on a foundation similar to the shipboard mounting for which it is intended. These requirements are recommended for new, replacement, or reworked equipment.

The SNAME T&R Code C-5, "Acceptable Vibration of Marine Steam and Heavy-Duty Gas Turbine Main and Auxiliary Machinery Plants," provides maximum allowable vibration levels for shop test and shipboard test as follows:

Figure 5-6 For steam turbine bearing housing or gear casing measurements

Figure 5-7 For gas turbine housing measurements

Figure 5-8 For steam turbine shaft measurements

These limits are narrowband readings of first order (rotational frequency) and second order vibration and apply to steady state operation, preferably under trial conditions called for under SNAME Code C-1 or ISO 4867. Measurements which exceed the limits called for by "Shipboard Test" indicate corrective action required.

In like manner, Figures 5-9 and 5-10 give the maximum acceptable levels for turbine driven auxiliaries for measurements made on bearing housing or shaft, respectively.

For motor-driven auxiliaries, the maximum first order and second order bearing housing vibration velocities of the assembled driver and driven equipment is recommended to be ± 0.25 inches per second above 30 Hz and ± 2.5 mils below 30 Hz. For new or replacement equipment, the values shown by MIL-STD-167, Figure 5-5 should be used.

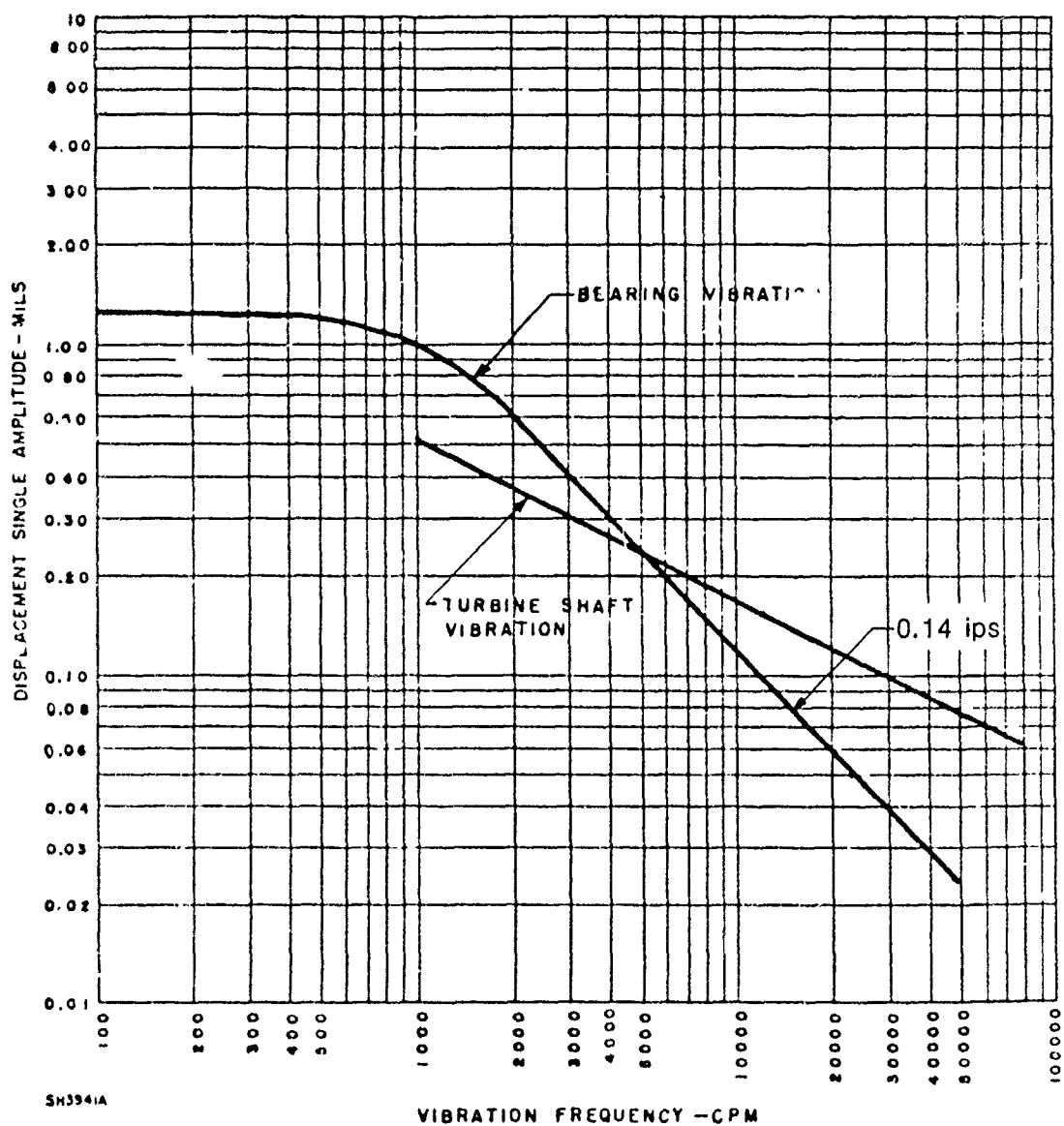


Figure 5-5. Maximum Allowable Vibration. Type II (MIL-STD-167-1 (SHIPS), 1 May 1974)

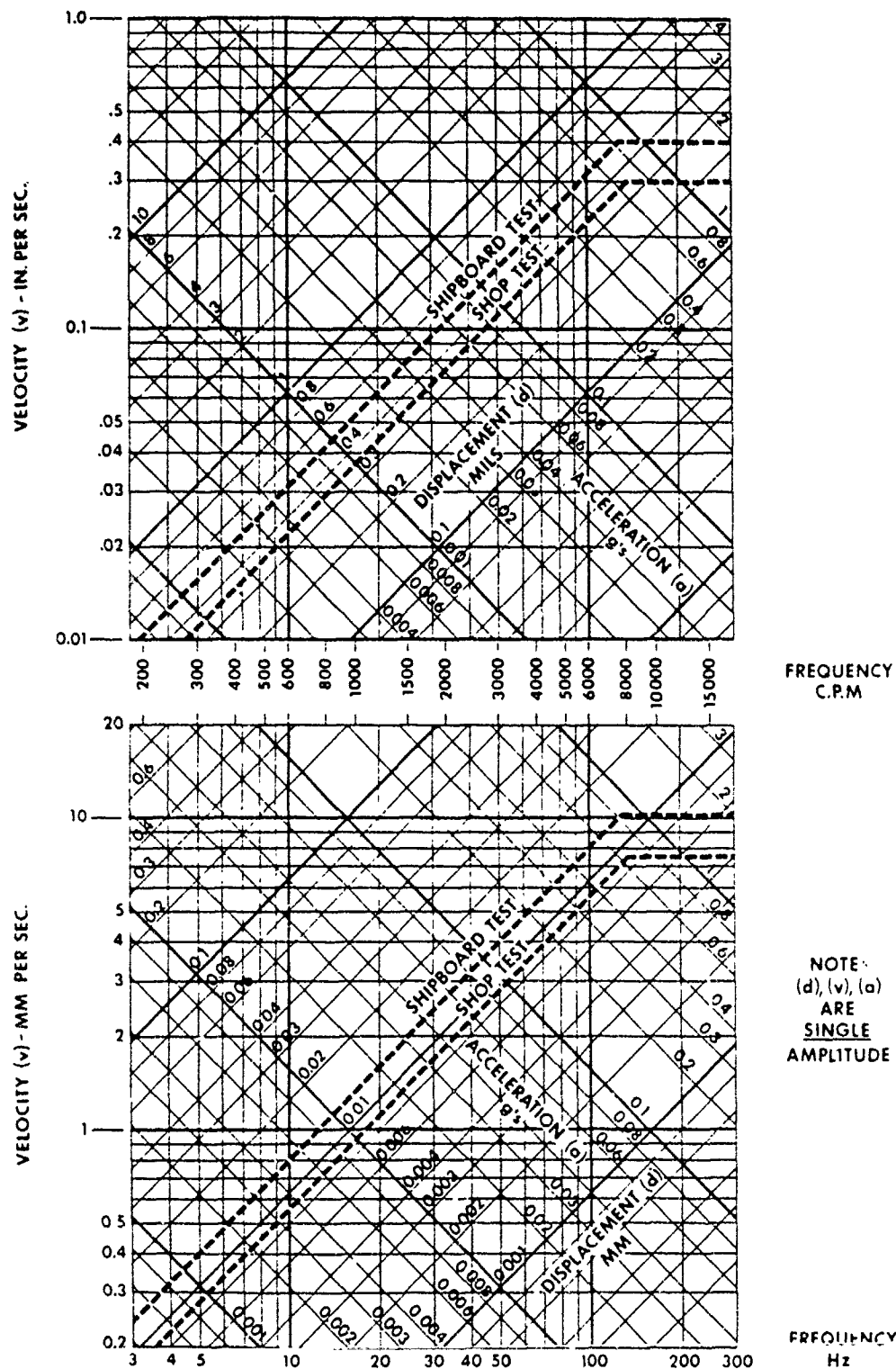


Figure 5-6. Main Propulsion Steam Turbines and Reduction Gears--Turbine Bearing Housing and Gear Casing Vibration Limits

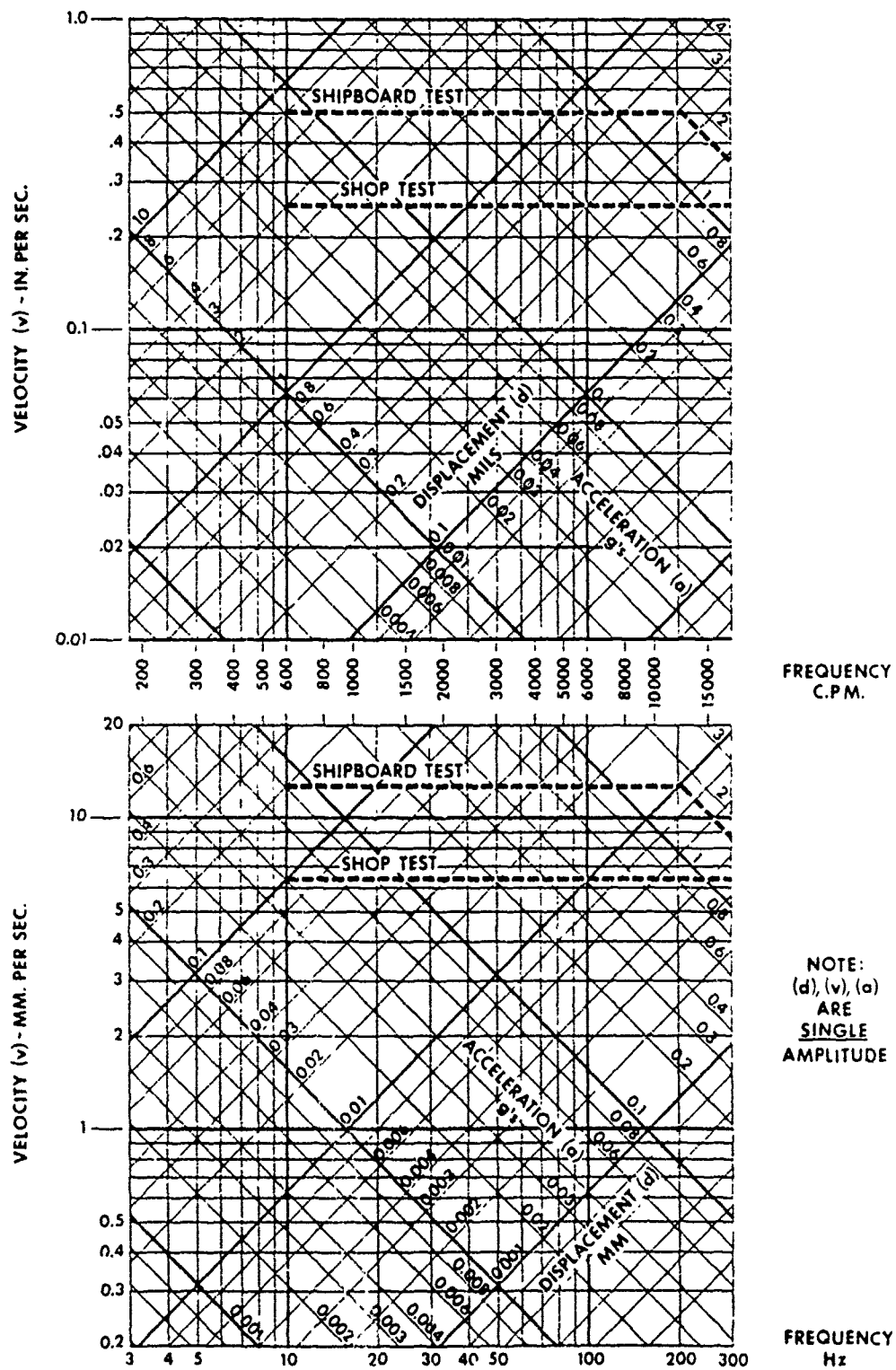


Figure 5-7. Main Propulsion Gas Turbines Bearing Housing Vibration Limits

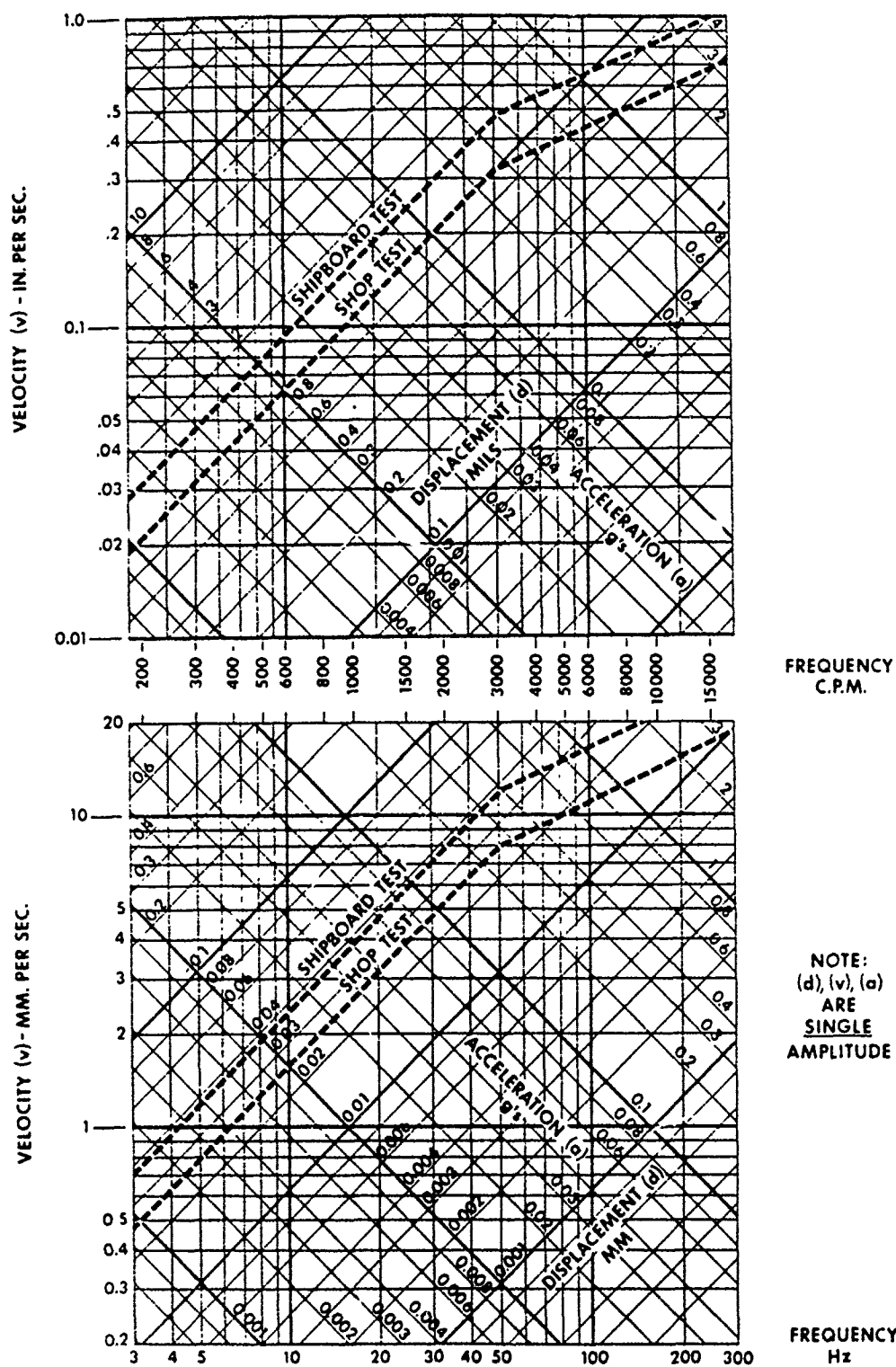


Figure 5-8. Main Propulsion Units--Shaft Vibration Limits

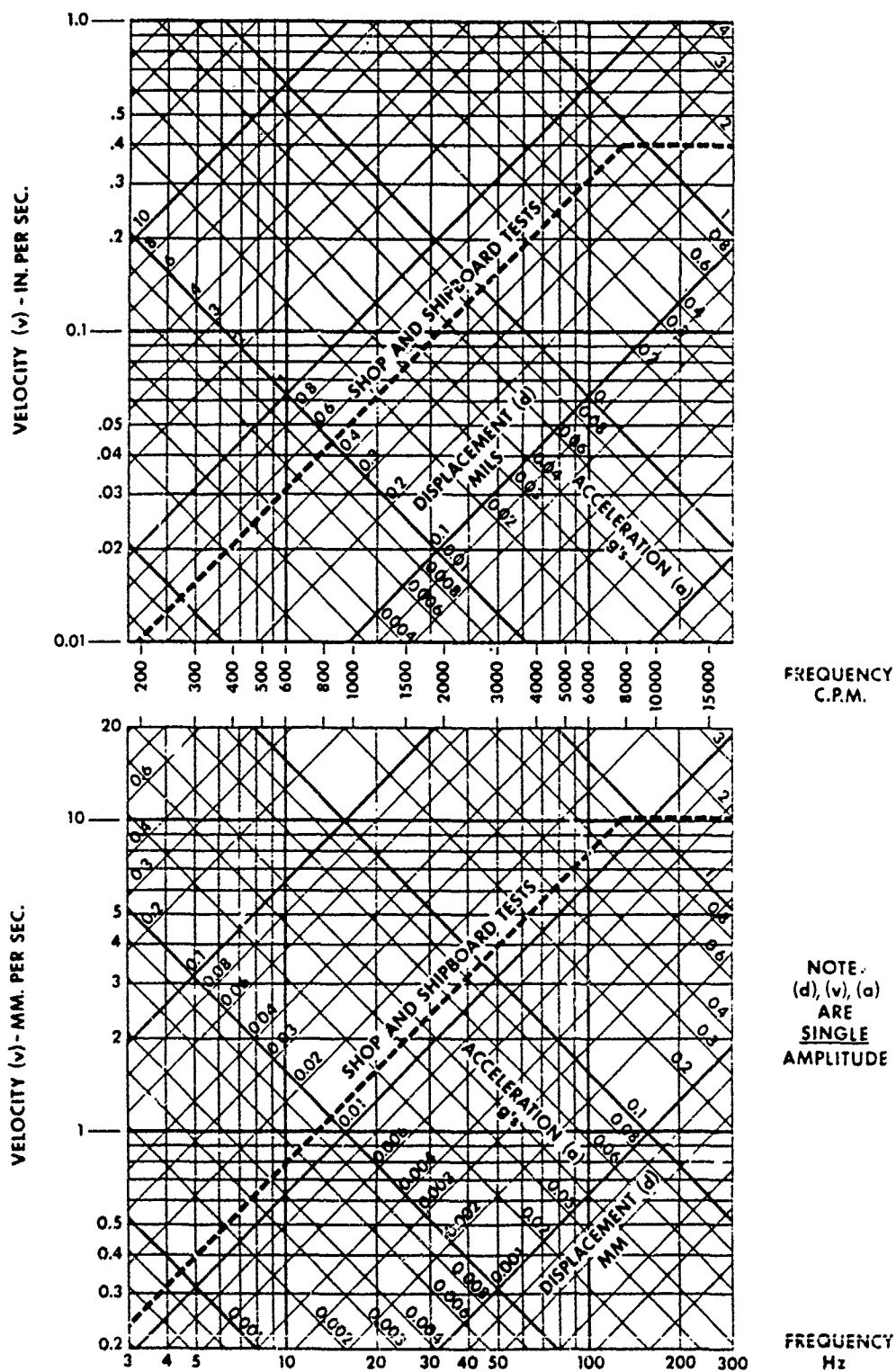


Figure 5-9. Turbine Driven Auxiliaries--Bearing Housing Vibration Limits

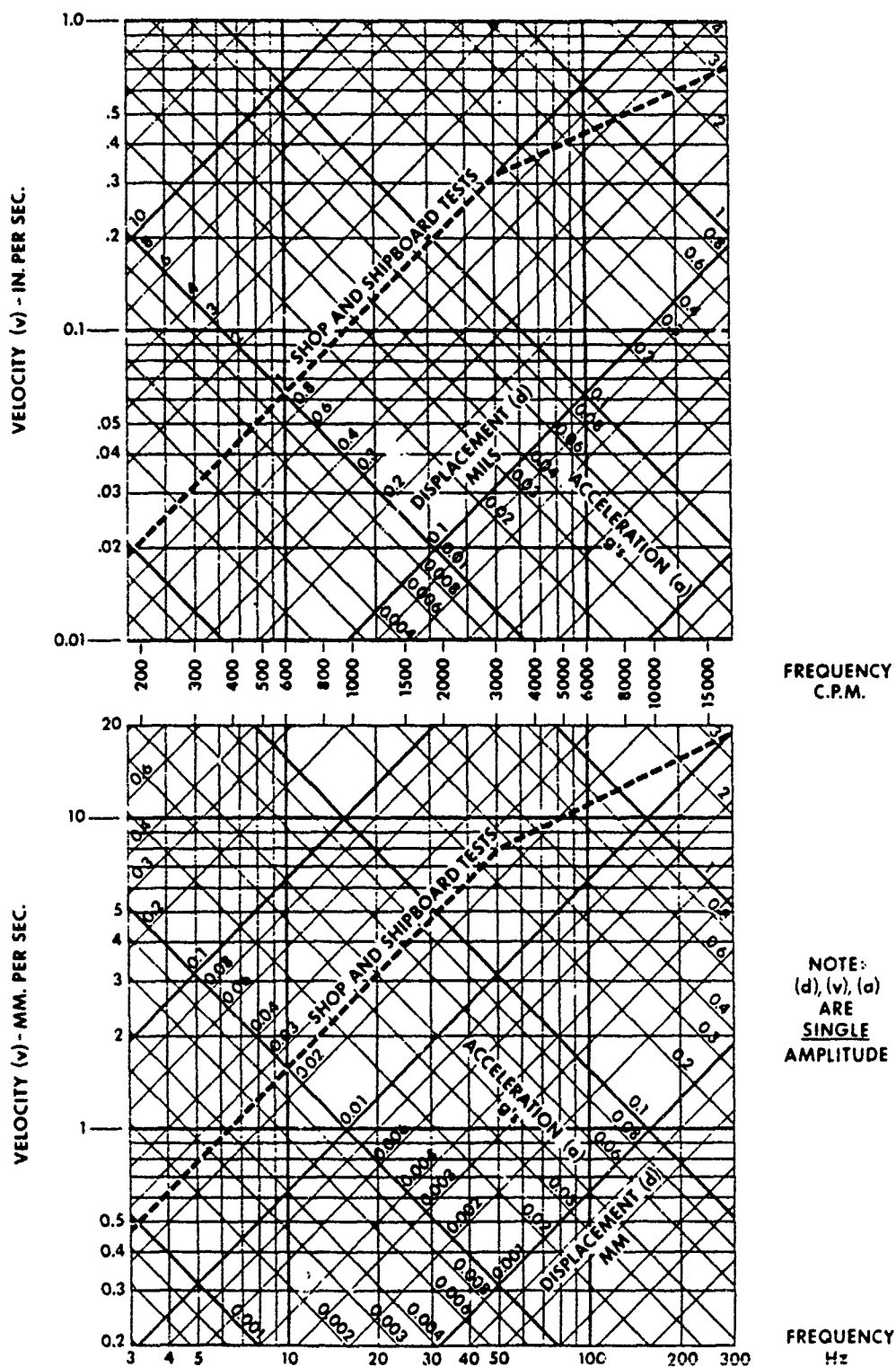


Figure 5-10. Turbine Driven Auxiliaries--Shaft Vibration Limits

5.3.2.2 Reciprocating Engines

Based on data presented by Bureau Veritas Guidance Note NI 1381-RD3, "Recommendations Designed to Limit the Effects of Vibration Onboard Ships," June 1979, the acceptable levels of diesel engines and reciprocating engines are as shown on Figure 5-11. Vibratory levels at ± 11 mm/sec measured at the base of the engines should be monitored, while ± 18 mm/sec for the smaller engines (< 1000 HP) and ± 28 mm/sec for larger engines (> 1000 HP) would be considered excessive. Somewhat higher levels could be tolerated at the cylinder heads.

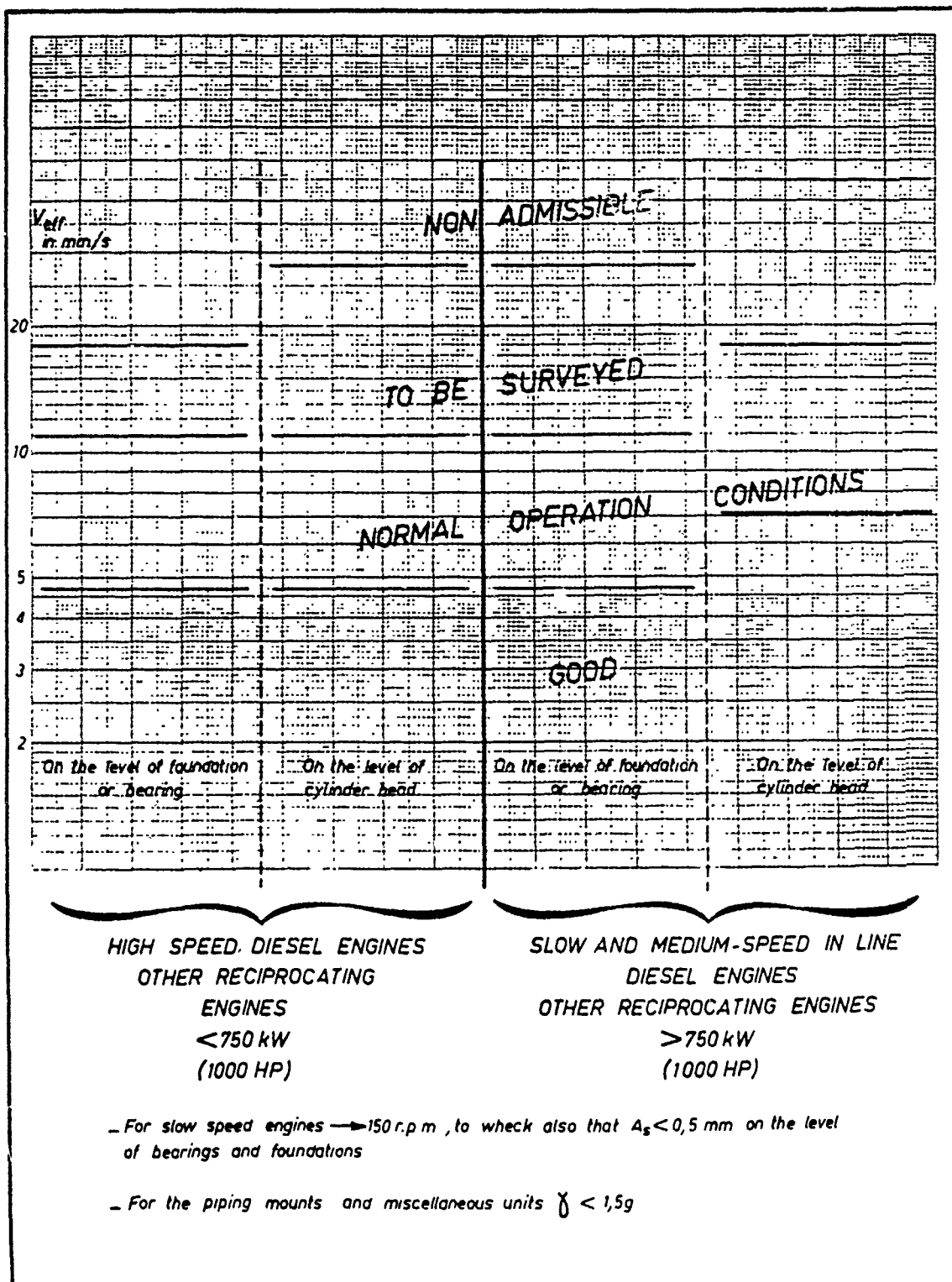


Figure 5-11. Vibratory Levels of Diesel Engines and Reciprocating Engines

SECTION 6.0

VIBRATION EVALUATION AND CORRECTIVE ACTION

This section will discuss the situation where a high level of vibration is known to exist, and it must be decided what to do to fix it, what additional data must be gathered, or whether to consult with a vibration engineer. It is helpful to understand the nature of the vibration and the forces causing it when making such a decision.

6.1 GENERAL

Normally, it is helpful to identify the mass(es), spring(s), damper(s), and force(s) in a vibrating system. If the mechanical system approximates the one degree-of-freedom system discussed in Appendix A, it is possible to anticipate the effect on the natural frequencies of varying the mass or stiffness of the system. If the vibrating system involves more than one mass, it usually is wise to calculate the effect of proposed changes. Also, with multiple degree-of-freedom systems, it may be necessary to take comprehensive data to define the motion adequately. Generally, multiple degree-of-freedom systems should be left to vibration engineers.

In evaluating a vibration problem, care must be given to attach the proper amount of importance to the symptoms. For example, often noise can be very obvious and annoying, but may be nothing more than dishes or loose gear rattling in response to normal levels of vibration. Other noises, such as rumbling in a reduction gear, or a squeal in a bearing can be very revealing and important. The same is true of vibrations that can be seen or felt. An amplitude of vibration large enough to be seen may not be serious if it involves a lightweight panel or a handrail. However, if it involves a rotating machine or a major structural member, it could be very serious.

In the preceding section various criteria for shipboard vibration were discussed. These should be referred to for the purpose of determining whether or not the levels of vibration measured or observed are too high. This section will present information to assist in understanding how the system is behaving and discuss various corrective actions that might be considered for hull, local, and machinery vibration.

6.2 HULL VIBRATION

High levels of hull vibration may be detected by routine measurements during sea trials. They will probably be stern vibration measurements, and they can be compared to levels in Section 5, Criteria, to determine acceptability. Crew discomfort or equipment damage may also indicate excessive levels of vibration. If discomfort was the symptom, the general magnitude of the vibration can be estimated from the chart in Section 5, depending on whether it is "annoying", "very uncomfortable", etc. In any case, this section will give clues as to the causes, and discuss possible solutions. In most cases, the cost

and effort involved in correcting a hull vibration problem makes it worthwhile to consult a vibration engineer, if only to confirm the crew's diagnosis.

6.2.1 Excitation

The first task is to identify the source of the exciting forces. This is primarily done by finding the frequency(ies) of the vibration. The following list gives the characteristics of the forces likely to cause hull vibration:

- Poor Flow Into the Propeller - Blade frequency forces are applied to the propeller shaft in the form of longitudinal, torsional, and lateral forces; and to the hull through pressure forces near the propeller and through the main shaft bearings. If high forces are caused by poor flow into the propeller, they will be evident primarily in the shaft vibration, and may or may not involve significant forces at blade frequency harmonics as well as blade frequency itself. Several techniques for correcting poor flow are:

- Install fins ahead of propeller to improve flow
- Use skewed propeller blades
- Increase propeller axial or radial clearances

Each of these techniques should be carefully analyzed by a vibration engineer before they are attempted.

- Propeller Cavitation - The propeller may cavitate because of a poor design, or because of a poor flow into the propeller. In either case, the forces are evident primarily on the hull, rather than the shaft, and they are accompanied by large forces at several blade harmonics as well as blade frequency. Also, inside the hull it will sound like a loud hammering on the hull near the propeller. The techniques mentioned above for improving flow or increasing clearances may be applicable. If the propeller is at fault, a new design or modification may be necessary. Temporarily, a small reduction in speed usually causes a large reduction in cavitation.
- Propeller Unbalance - Mass unbalance will cause lateral vibration of the shaft, and thence the hull, at shaft frequency. If the propeller is damaged, mass unbalance will probably be accompanied by pitch unbalance, which will increase the axial, torsional, and lateral forces on the shaft at shaft and probably twice shaft frequencies. To correct this, the propeller has to be replaced or repaired, including balancing.

- Sea Forces - Interaction of the ship with the sea causes vibration by means of several mechanisms. If the bow lifts out of the water and slaps as it re-enters (slamming), it excites the vertical hull modes. This is a transient phenomenon and usually occurs only occasionally. Normally, the hull forward is flat on bottom, causing very high slamming impact loads. If the hull form could be changed to a round or a vee shape, it would help tremendously. Otherwise, a course change and/or speed reduction are about the only recourses. Wave slap, without bow emergence, and green water on deck are two other transient excitations of the same nature, but usually not as serious. Sometimes, in moderate seas, ships with a very low fundamental frequency, say 1/3 to 1/2 Hz, will react to the seaway with large (several inches) vibrations. This is called "springing", is most common on large Great Lakes ore carriers, where 12" amplitudes are not unusual, and occurs because the natural frequency is within the frequency range of the wave energy. Some improvement would result from refining the bow and stern lines of the ship to reduce the forces. Course and speed changes, or redistributing the load are the usual solutions but, since wave energy covers a range of frequencies, a large change may be required.

6.2.2 Location of Vibration

The extent of the high vibration levels is another factor in evaluating vibration problems. If the hull vibration appears only in the stern area, in the case of blade frequency, that is an indication that the propeller forces are excessive, rather than normal forces being magnified by hull resonances. If blade frequency coincides with a hull resonance, the high levels may be apparent over the length of the hull, even though the excitation forces are not excessive. Similarly, resonances of the deckhouse, a deck, a mast, etc., may be excited by normal propeller forces. If a resonance is known to exist, the stiffness of the structure can be changed, or the exciting frequency can be changed by changing the number of blades or shaft speed. If there are no practical methods of changing either the structural frequency or the excitation frequency, one of the methods for reducing propeller forces could be considered.

6.2.3 Hull Vibration Troubleshooting Table

Table 6-1 summarizes the evaluation and correction of hull vibration problems in the form of a troubleshooting table.

6.3 LOCAL VIBRATION

In addition to hull girder vibration problems, there may be problems with major substructures such as the deckhouse, mast, propulsion system, or

Table 6-1. Hull Vibration Troubleshooting Table

| <u>LOCATION</u> | <u>CAUSE</u> | <u>FREQUENCY</u> | <u>DIRECTION</u> | <u>CORRECTION</u> |
|---|--|-------------------------------|---|---|
| STERN AND AFT DECKHOUSE | POOR FLOW TO PROP | BLADE, SOME HARMONICS | ALL | INCREASE PROP CLEARANCES, INSTALL FINS, CHANGE PROP, SKEW BLADES |
| STERN AND AFT DECKHOUSE | CAVITATION | BLADE, STRONG HARMONICS | ALL | SAME AS ABOVE, PARTIAL TUNNEL OVER PROPELLER |
| ENTIRE LENGTH OF HULL (EXCEPT AT NODES) | RESONANCE OF HULL WITH PROP FORCES | SHAFT OR BLADE | VERTICAL OR ATHWARTSHIP | CHG NO. OF PROP BLADES, STIFFEN HULL, CHG SHAFT SPD |
| STERN OR ENTIRE HULL (SHAFT) | PROP MASS UNBALANCE | SHAFT | VERTICAL OR ATHWARTSHIP (LATERAL) | BALANCE PROP |
| STERN OR ENTIRE HULL (SHAFT) | PROP PITCH UNBALANCE (BENT BLADE) | SHAFT, 2 X SHAFT | VERTICAL OR ATHWARTSHIP (AXIAL, TORSIONAL, AND LATERAL) | REPAIR OR REPLACE PROP (AND BALANCE) |
| ENTIRE HULL (ROUGH SEA) | SLAMMING | LOW HULL MODES | VERTICAL, CHG COURSE, | REDUCE SPD, ELIMINATE FLAT BOTTOM |
| ENTIRE HULL | WAVE SLAP, GREEN WATER ON DECK (ROUGH SEA) | LOW HULL MODES | VERTICAL, SOME ATHW | REDUCE SPD, CHG COURSE |
| ENTIRE HULL | SPRINGING (MODERATE SEA) | 1ST HULL MODE (1/3 TO 1/2 HZ) | VERTICAL (SOMETIMES HORIZ. OR TORSIONAL) | REFINE BOW AND STERN LINES, CHG COURSE, SPEED, OR LOADING |
| ENTIRE SUB-STRUCTURE | DECKHOUSE OR DECK RESONANCE W/PROP FORCES | USUALLY BLADE OR 2 X BLADE | VERTICAL, ATHW, OR LONGITUDINAL | STIFFEN STRUCTURE OR AVOID RESONANT SPEEDS |
| ENTIRE SUB-STRUCTURE | HULL, DECK OR DECKHOUSE RESONANCE W/ ENGINE FORCES | ENGINE REV. OR HARMONICS | VERTICAL, ATHW, OR LONGITUDINAL | STIFFEN STRUCTURE, CHANGE ENGINE MOUNTS, OR AVOID RESONANT SPEEDS |

large portions of the deck structure. Such large sections of the ship, when they vibrate, will affect the vibration of the hull girder itself. In an analysis, the hull model should include a representation of that structure. One rule of thumb is that if the mass of a substructure is greater than 1/2 percent of the displacement of the ship and has a natural frequency close to a hull frequency, it should be represented as separate from but attached to the hull girder.

The term "local vibration" usually refers to the vibration of smaller structures or pieces of equipment, small enough so that its vibration will not significantly effect the hull girder vibration. Examples include:

- Plates and panels - Bulkheads, webframes, deck or bottom sections
- Beams - Shafting, masts, cranes, antennas, pipes
- Plate and beam assemblies - section of deck, bottom, bulkhead, shell, or superstructure
- Equipment mounted on elastic foundation - boilers, condensers, turbines, auxiliary machinery, electronic equipment (may be mounted on resilient mounts or hard mounted to structure)
- Appendages - rudders, skegs, roll fins, propeller blades

6.3.1 Excitation of Local Structures

The sources of excitation for local structures include all of the sources mentioned for hull girder excitation: propeller, engines, and waves. These excite local structures through the hull girder motion. Even if the hull motion is acceptable, a local structure mounted to it may resonate at one of the frequencies of excitation. In addition, auxiliary equipment may excite itself or nearby structures or equipment. In the case of appendages, the flow of water past the appendage and its interaction with the structure can excite vibration.

The frequency and magnitude of the excitation through the hull girder will vary with the location. In the after peak area, strong blade, 2 x blade, and 3 x blade would be expected. The more cavitation there is the stronger the harmonics will be. In the engine room area the main orders of the engine would be expected. In the superstructure and in the hull in general, the effects of both the propeller and engines will be present, but the higher harmonics or orders will fade fastest as the distance from these sources increases. On the bow there will probably be very little evidence of either source unless a hull mode coincided with one of the excitations.

The forces from the sea will be applied primarily to the bow and will be impulsive in nature. The hull will respond primarily in its lower modes.

A local structure may respond differently if it is located in one part of the ship than another because of the mode shape of the hull. This is somewhat true of all hull girder excited local vibrations, but especially true when slamming, wave slap, and springing are present. The best example of this might be a mast. If it were located in the middle of the ship, the first hull mode would provide a vertical motion of the base, but no rotation. This is unlikely to cause a problem. If it were located near one of the hull nodes, say one quarter of the length from the bow, the hull would provide a rotation of the mast base, but little translation. If the natural frequency of the mast was the same as the hull, severe whipping of the mast is likely.

Hydrodynamic forces applied to the appendages may be the result of Karman vortices or hydrodynamic flutter. Karman vortices are caused by an appendage with a blunt trailing edge. The low pressure behind the trailing edge causes eddies to form. The normal laminar flow is likely to collapse on one side or the other, not both simultaneously. When it does collapse it relieves some of the suction, and the other side maintains its laminar flow. As the eddy trails downstream, and the collapsed side smooths out again, the rotation of the eddy provides enough differential in pressure that the opposite side now collapses into an eddy. This alternate pattern is maintained, causing an alternating pressure on the trailing edge of the appendage. If the frequency of the vortices coincide with a natural frequency of the appendage, vibration will result. For a particular trailing edge geometry, Karman vortices are likely to form only for a range of water velocities. Below that range the flow is laminar and above the range it is completely turbulent. The vortex frequency decreases with the bluntness of the trailing edge. The most common shipboard occurrence of this phenomenon is the "singing propeller," where the vibration of the propeller blade causes an audible tone that may be heard (sometimes to the point of distraction) inside the ship.

Karman vortices can also be shed from masts and stacks at certain wind speeds, and if a natural frequency of the structure coincides with the vortex shedding frequency, severe vibration can result.

Hydrodynamic flutter is an interaction of a vibrating appendage and the flow across it. A rudder for example may be excited by a horizontal vibration of the ship's stern. If the hydrodynamic and inertial forces are not balanced, the rudder will rotate slightly as well translate. The rotation increases the "lift" on the rudder and causes it to deflect more, aggravating the vibration. The likelihood of this happening, and how serious it is, will depend on the frequency of excitation, the natural frequencies of the rudder (lateral and torsional), and how the lift forces are applied to the rudder.

6.3.2 Evaluation

Local vibration, like hull vibration, will be judged unacceptable if it causes crew discomfort, fatigue of structures, or equipment failure. Section 5, Criteria, discusses the norms against which vibration levels in general can be judged. Often, however, a problem must be considered and judged individually. This section will discuss some guidelines for such evaluations.

The fatigue of structures is difficult to assess with normal vibration measurements. Fatigue damage is related to the stresses in the material and the number of cycles of stress experienced. The stresses are proportional to vibratory displacement, but the relationship between the two is not easily determined. Stress concentrations, such as inside corners and holes, are particularly hard to evaluate and, of course, that is where damage is likely to occur first. If the stress level is known it should be compared to the fatigue limit of the material. If the number of cycles is known to be less than 1,000,000, the material may survive stress levels above the fatigue limit. In this case, reference should be made to a S-N curve (Stress vs No. of Cycles to Failure) for the material in question. In any case, a conservative approach is to keep the stress levels below the fatigue limit by some factor of safety, usually around two for machined parts. The factor of safety for castings should be much higher. A rule of thumb for most ship structures is that fatigue problems are not likely if vibration displacements are below \pm or $-$ 1 mm (about .040") and velocities are below \pm or $-$ 100 mm/sec (about 4"/sec).

The natural frequency of the local structure is probably the most important characteristic in evaluating a problem. Generally, if the natural frequency of a structure is about 15 percent above or below an exciting frequency, there will be no problem. The natural frequency may be known from measurements while the ship is underway or a machine is operating. Another method for finding the natural frequency is to excite it artificially and measure the resulting vibration. Impacting the object with a hammer (usually a rubber or leather hammer is preferred to minimize high frequency effects) will cause it to "ring" at its natural frequencies. Large structures may require a heavy timber to get sufficient excitation. A more controlled method is to use a vibration generator to provide a sinusoidal force at various frequencies. The weight of the vibration generator should not be so much that it affects the vibration of the object.

If stiffening or some other fix is anticipated, it is often a good idea to find the existing natural frequency, compare it to the calculated natural frequency to see how good the model is, and then calculate the frequency of the object with the fix. Often changes in the structure have less effect than anticipated. It is also prudent to test the modified structure in order to verify the affected fix.

The formulas in Appendix A can be used for calculations. Many local problems will involve panels. Most panels on ships (decks, bulkheads, etc.) will have natural frequencies somewhere between the simply supported and the clamped cases.

6.3.3 Corrective Techniques

A major part of "correcting" vibration problems consists of redoing something that was not done properly in the design of the ship. Therefore, it is often appropriate to review certain design guidelines to see if they apply. The following may be helpful:

- Make vertical bulkheads (longitudinal and transverse) as continuous as possible from deck to deck. Where this is difficult, perhaps stanchions can be used for continuity.
- Make sure that heavy pieces of equipment are installed over beams, bulkheads, frames, or webs, preferably in both directions (longitudinal and transverse).
- Avoid cantilevers to support equipment (unless designed with vibration in mind).
- Make bearing foundations sturdy.
- Where large forces are involved between machinery and foundations (engines, turbines, gears, thrust bearing) the manufacturers should be consulted on the stiffness of foundations.
- Double bottoms should be deep, and should change depth gradually if a lesser or greater height is needed to accommodate machinery or equipment.

More often than not the solution to local vibration problems is to increase stiffness. Figure 6-1 shows several examples and some possible solutions.

In Figure 6-1 (a), a piece of equipment is mounted on a foundation, the legs of which do not rest on the supporting beams of the deck. The deck flexes and the equipment rocks or vibrates vertically. The solution could include moving the equipment so the legs rest on the beams, adding additional legs that rest on the beams, or adding a cross stiffener (above or below the deck).

In Figure 6-1 (b), the legs of the foundation are too flexible. This can be corrected by installing diagonal braces between the legs or outside of them, or attaching plates across the legs and from deck to platform.

In Figure 6-1 (c), a panel vibrates too much and needs to be stiffened. The natural frequency is controlled by both dimensions, but primarily by the shortest span. In stiffening the panel, an effort should be made to decrease the shortest span, such as shown in the last two solutions.

6.4 MACHINERY VIBRATION

The most important benefits that a ship's crew can realize from their own efforts in the area of vibration have to do with machinery. Relatively simple measurements can be used to monitor the mechanical condition of pumps, generators, turbines, motors, etc. Also, with a little bit of training and simple equipment, a limited amount of diagnostic work is possible. However, any

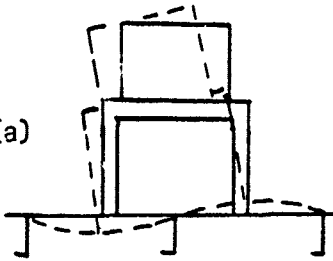
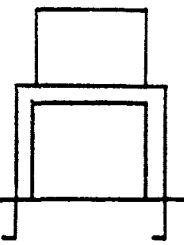
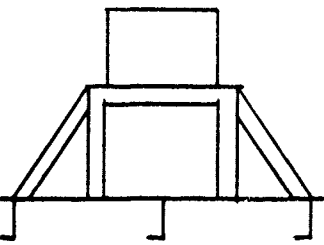
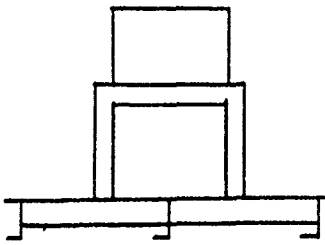
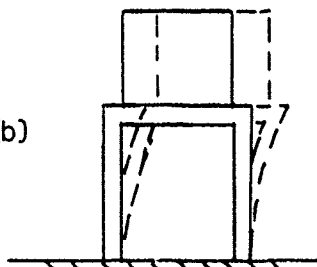
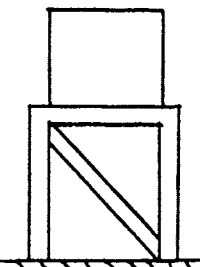
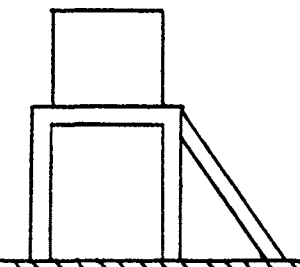
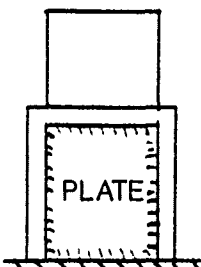
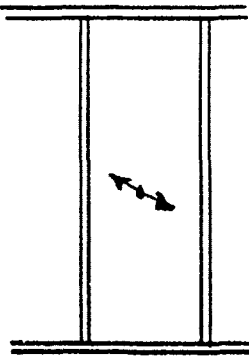
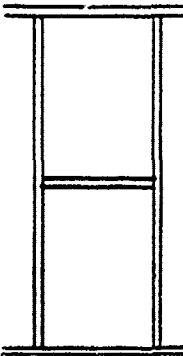
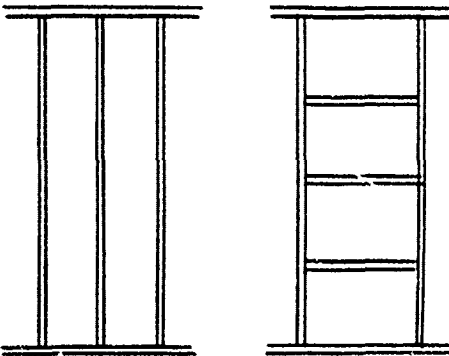
| PROBLEM | SOLUTIONS | | |
|--|---|---|---|
| <p>(a)</p>  <p>FLEXIBLE DECK</p> |  <p>MOVE</p> |  <p>BRACE FROM BEAMS</p> |  <p>STIFFEN DECK</p> |
| <p>(b)</p>  <p>FLEXIBLE FOUNDATION</p> |  <p>INTERNAL BRACE</p> |  <p>EXTERNAL BRACE</p> |  <p>PLATE</p> |
| <p>(c)</p>  <p>PANEL</p> |  <p>NOT VERY EFFECTIVE</p> |  <p>MUCH BETTER (REDUCE SHORT SPAN)</p> | |

Figure 6-1. Solutions to Local Vibration Problems

problems involving main propulsion machinery should be left to vibration engineers. This section discusses all these topics, but emphasizes the topics that the crew should be primarily involved with: diagnostics of rotating machinery and vibration monitoring.

6.4.1 Rotating Machinery

In order to identify the cause of high levels of vibration in rotating machinery, the frequency, amplitude, and direction of vibration can be used as indicators. Table 6-2 is a troubleshooting chart for machinery.

6.4.2 Machinery Monitoring

Vibration measurements on machinery is being used increasingly on a periodic basis to detect degradation before it reaches the point where a machine has to be shut down. This enables the crew to schedule maintenance operations rather than perform maintenance on an emergency basis. Often vibration measurements can substitute for more time-consuming disassembly and inspection.

Machinery monitoring measurements are often taken each month, so it generally requires onboard test equipment and execution by the crew. Experience with particular machinery will suggest whether to make measurements more or less often.

Measurements can be taken with very simple instruments, or very sophisticated ones. Most often, they will involve one of the following types of system:

- Transducer with Meter
- Transducer with Filter and Meter
- Transducer with Spectral Analyzer

Each of these will be discussed in this section as the need arises.

No matter what instruments are being used, the first step is to obtain "baseline" measurements. Measurement locations must be selected for each machine in the maintenance program. The locations mentioned in Section 6.4.6 can be used for guidance. Then the vibration of each location on each machine is measured while the machine is in good operating condition. These are the "baseline" vibration levels and will be compared to measurements made periodically at the same locations. Measurements should also be made after a machine has been overhauled.

General guidelines can be given in interpreting the vibration data. Both the absolute level of vibration and the change from the baseline are important. If the absolute level exceeds the levels of safe operation given in Section 5.0, the machine should be shut down. If the vibration increases by a factor of two over the baseline measurement, there may not be a problem, but

Table 6-2. Machinery Troubleshooting Chart

| <u>CAUSE</u> | <u>FREQUENCY</u> | <u>DIRECTION/ AMPLITUDE</u> | <u>REMARKS</u> |
|--|---|-------------------------------------|--|
| UNBALANCE | 1 X RPM | RADIAL/STEADY | MOST COMMON CAUSE |
| BENT SHAFT | 1 OR 2 X RPM | AXIAL/HIGH | |
| ANTI-FRICTION BEARINGS | 20 TO 50 X RPM | RADIAL/LOW, UNSTEADY | AS FAILURE NEARS, AMPLITUDE INCREASES FREQUENCY DECREASES |
| SLEEVE BEARINGS | 1 X RPM | NOT LARGE | SHAFT AND BEARING AMPLITUDES ABOUT THE SAME |
| COUPLING OR BEARING MISALIGNMENT | 2 X RPM (SOMETIMES 1 OR 3 X RPM) | HIGH AXIAL (CAN BE TWICE RADIAL) | |
| DEFECTIVE GEARS | RPM X NO. OF GEAR TEETH | RADIAL/LOW | |
| MECHANICAL LOOSENESS | 1 OR 2 X RPM | RADIAL | VIBRATION LARGEST IN DIRECTION OF LOOSENESS |
| DEFECTIVE BELT | 1 OR 2 OR MORE X BELT RPM | RADIAL/UNSTEADY | |
| ELECTRICAL | 1 OR 2 X PWR FREQUENCY (60 OR 120 HZ) | USUALLY LOW | VIBRATION STOPS IF PWR IS TURNED OFF |
| OIL WHIP | LESS THAN RPM | RADIAL/UNSTEADY | FREQUENCY MAY BE AS LOW AS 1/2 X RPM |
| AERODYNAMIC OR HYDRODYNAMIC | 1 X RPM OR NO. OF BLADES X RPM | RADIAL OR AXIAL | MAY BE PROBLEM IF RESONANCE EXISTS |
| BEAT FREQUENCY | 1 X RPM | PULSATING AT BEAT RATE | CAUSED BY TWO MACHINES WITH CLOSE RPM |
| RESONANCE | SPECIFIC CRITICALS | HIGH IN DIRECTION OF RESONANCE | PHASE WILL CHANGE WITH SPEED. AMPLITUDE WILL DECREASE ABOVE AND BELOW RESONANT RPM |
| HYSTERESIS WHIRL | SHAFT CRITICAL SPEED | PRIMARILY RADIAL | VIBRATION EXCITED WHEN PASSING THRU CRITICAL IS MAINTAINED AT HIGHER SPEEDS |

measurements should be made more often, perhaps once a week, to see if the vibration continues to increase or if it stabilizes. If it increases by a factor of 10 over the baseline, it should be repaired, even if it does not exceed the levels in Section 4. If there are two identical machines, comparing their vibration levels is usually helpful.

So far the discussion has been about whether or not there is a problem, with no insight into what the nature of the problem might be. How the machine is diagnosed depends on the type of measurement equipment being used.

If only a transducer and meter are used, there is no indication of the frequency of the increased vibration, and the measurement does not help much in the diagnosis. An exception is a vibration meter that will read displacement and velocity, velocity and acceleration, or all three. If the displacement increases more than the velocity or acceleration, the frequency is low, such as it would be with an unbalance problem. If the acceleration increases more than the velocity or displacement, the frequency is high, such as a bad bearing might cause. A list of common problems and the associated frequency range is as follows (Reference 6-1):

Low Frequency

- Unbalanced rotor - worn, eroded, broken, or corroded parts
- Misalignment - induces significant axial vibration
- Eccentric shafts
- Slipping clutches
- Mechanical looseness
- Loose foundation bolts
- Oil Whirl or slip - one-half or less times shaft speed
- Worn belts
- Belts and pulleys out of adjustment
- Changed reciprocating elements that introduced additional torsional vibration

High Frequency

- Vibration due to impacts
- Defective bearings - random or rough vibration
- Poor gears
- Slipping clutches
- Rubbing or binding parts
- Air leaks
- Hydraulic leaks
- Parts colliding
- Broken or loose pieces
- Electromagnetically driven loose pieces
- Water hammer
- Surge

If a filter is available to use with a transducer and meter, the frequency as well as amplitude of the major components can be found. Refer to the troubleshooting table in Section 6.4.1. Some thought should be given to how much use of the filter is worthwhile, since it is time-consuming to record all major components of all measurement locations. As a minimum, the filter can be used to diagnose a problem after it has been detected by measuring excessive unfiltered vibration. It may be worthwhile to measure all major components for the baseline measurements and, then again, only for diagnosis. In that case, the unfiltered vibration should also be measured for the baseline and compared to the unfiltered periodic measurements. This would seem to be a reasonable compromise.

If the amplitudes of the individual frequency components are desired for the periodic measurements, the use of a filter would probably not be worthwhile. The same data can be obtained much faster by means of a spectral analyzer. Some of these are reasonable in cost, and produce a hard copy plot of the frequency spectra for each location. They are easy to compare with the baseline spectra, and are convenient to file in the machinery history.

6.4.3 Externally Excited Machinery

Occasionally a machine will vibrate excessively as a result of its supporting structure being excited by some other source, such as another machine, propeller forces, etc. The source can perhaps be identified by its frequency. Also, the vibration can be observed while nearby machinery is operated in turn, or while underway and in port. In any case, such externally excited machinery can usually be treated in the same manner as a local structure or equipment problem. The reader is referred to Section 6.3.

6.5 MAIN PROPULSION MACHINERY VIBRATION

The subject of main propulsion machinery vibration could fill several volumes all by itself. It includes the longitudinal, lateral, and torsional vibration of the propeller shafting system and the vibration of the prime mover, whether it be a turbine system or diesels. Each of these subjects has its own history of development, problems, analytical techniques, design criteria, and prospective fixes. Each of these subjects is treated briefly, but it is recommended that problems in these areas be referred to vibration engineers.

6.5.1 Longitudinal Shaft Vibration

Longitudinal shaft vibration is caused by blade frequency alternating thrust on the propeller forcing the entire propeller, shaft, thrust bearing and reduction gear assembly, and perhaps more, to oscillate fore and aft. In conventional designs, the fundamental natural frequency is often in or near the operating range of blade frequency, causing very high amplitudes.

If the natural frequency is in the lower power range, a resonance may not result in excessive vibration, but if it is near full power, it almost always will be damaging. Possible consequences are:

- Thrust reversal, particularly in turns, damaging thrust bearing
- Excessive stresses on gear teeth in reduction gear
- Excessive motion across couplings between gear and engine or turbine, causing wear
- Excessive deflection of attached piping or structure, causing failure
- Excessive motion of turbine rotor or engine crankshaft

Before corrective action can be taken, the location and severity of the resonance should be known, as well as the magnitude of the exciting force. If the blade frequency forces are excessive, it may be due to a highly disturbed flow into the propeller or poor propeller design. Fins have been installed on some ships just ahead of the propeller to improve the flow. A redesigned propeller may be required, possibly with skewed blades to reduce the forces.

If the resonance occurs near full power, there are several well known fixes, but each has its drawbacks. First, the number of blades on the propeller can be changed; increased to put the resonance at a lower speed where the power and forces are less, or decreased to put the resonance out of the operating speed range. The latter is usually not practical because a propeller with fewer blades normally has higher alternating forces. That fact, combined with the fact that there will be a significant magnification at full power even though operating on the flank of the resonance, makes this solution unreliable. Another fix is to move the resonance above the operating range. This can be done by moving the thrust bearing aft, or stiffening the thrust bearing housing and/or foundation. Moving the thrust bearing aft has the most effect, involves the most work, and complicates lubrication and maintenance. Stiffening the foundation sometimes is sufficient, but it usually requires a drastic amount of reinforcement to move the resonance a limited amount.

A rough approximation of the fundamental frequency can be made for most propulsion systems by means of a two-mass, two-spring model. Two natural frequencies can be calculated from the formula in Appendix A, Section A.3, but only the lowest would have any accuracy. The parameters, referring to Appendix A, Section A.3, are:

M_2 = Mass of propeller, including 60 percent for entrained water, plus 1/2 mass of shaft between propeller and bull gear (lb-sec²/in).

M_1 = Mass of propulsion machinery, including engines, turbines, reduction gears, casings, condensers, and 1/4 mass of shaft between propeller and bull gear ($\text{lb-sec}^2/\text{in}$). (If there is an axial coupling between the shaft and engine, $M_1 = 0$.)

K_1 = Spring constant of shaft and thrust bearing (lb/in).

K_2 = Effective fore-and-aft spring constant of foundations for the engine or turbine, gear, and thrust bearing (lb/in).

For any longitudinal problems, a vibration engineer should make a detailed analysis of the present and proposed systems before any fixes are attempted.

6.5.2 Torsional Vibration of Propulsion System

The rotating components of a propulsion system, including propeller, shaft, gears, and turbines or engines, can vibrate in torsional modes when excited by the propeller or engines. Turbines and gears in good mechanical condition do not significantly excite torsional vibration. It is usually all but impossible to avoid torsional blade frequency resonances in the entire operating range, so an effort is made in design to put resonant speeds where they will not cause problems, and to control mode shapes so the excitation points have minimum amplitudes. If the propeller, for example, were a node for a particular mode of vibration, that mode could not be excited by propeller forces.

Excessive torsional vibration may be evident from the following:

- Wear or rumbling in reduction gears
- Deterioration in couplings
- Fatigue failure of shaft or crankshaft

In general, torsional vibration is not felt or visually apparent at the bearings, casings, etc., enclosing the rotating parts. The exception is in reduction gears, where torsional forces become tangential forces on the gears, and cause bearing reactions.

Torsional vibration can be coupled with other modes of vibration. For example, the pitch of the propeller blades causes axial forces and longitudinal vibration in response to torsional oscillations.

On turbine driven ships, propeller blade frequency is the only excitation considered in design. Diesel engines excite torsional vibration at several orders and half orders (i.e., 1/2, 1, 1-1/2, 2, 2-1/2, etc.) as a result of both firing forces and reciprocating unbalance forces. In order to minimize and isolate these forces, diesel engines often have torsional dampers fitted to the crankshaft and/or torsional isolation couplings on the engine output shaft.

A rough estimate of the torsional fundamental frequency of a propulsion system can be found by considering it to be a shaft with a rotor on each end. The moment of inertia of the rotor on one end is that of the propeller plus an allowance for entrained water (usually 25 percent). The moment of inertia of the other rotor is that of the engine or turbines with reduction gears. The natural frequency of the system is

$$f_n = \frac{1}{2\pi} \sqrt{\frac{k_r(J_e + J_p)}{J_e J_p}}$$

where f_n = Natural frequency (Hz)

k_r = Spring constant of shaft (in-lbs)

J_e = Mass polar moment of inertia of engine, including gears (in-lb-sec²)

J_p = Mass polar moment of inertia of propeller, including water (in-lb-sec²)

The spring constant is

$$k_r = \frac{\pi G(D_o^4 - D_i^4)}{32L}$$

where D_o = Outside diameter of shaft (in)

D_i = Inside diameter of shaft (in)

G = Shear modulus of elasticity (12×10^6 lb/in² for steel)

L = Length of shaft (in)

The effect of the reduction gear is to make the inertias and stiffnesses of the higher speed elements effectively much greater. To account for this, use all the parameters calculated from the slow speed elements as they are, but multiply all the inertias and stiffnesses of the high-speed elements by the square of the gear ratio. If there is a double reduction gear, parameters for the intermediate speed elements must be multiplied by the square of the second reduction gear ratio, and parameters for the high-speed elements must be multiplied by the square of the product of the first and second gear ratios. The inertias of the propellers, gears, engines, etc., should be available from manufacturers data.

Geared systems usually have considerably larger inertias on the machinery end as compared to the propeller due to the magnifying effects of the gears. As a result, the fundamental torsional frequency may be estimated by considering the inertia at the machinery end to be infinite. The frequency is then:

$$f_n = \frac{1}{2\pi} \sqrt{\frac{k_r}{J_p}}$$

If indications suggest the presence of torsional vibration in the propulsion system, an experienced vibration engineer should be consulted.

6.5.3 Lateral Shaft Vibration

Lateral vibration of the shaft causes high forces on the line and stern tube bearings resulting in wear. In some cases, reduction gear wear or shaft fatigue could be a problem. Lateral shaft vibration can be caused by:

- Mass or pitch unbalance of propeller
- Mass unbalance of shaft
- Propeller bearing forces at blade frequency
- Bearing problems
- Bent shaft or misalignment at flanges

Note that misalignment of bearings is not included. This normally forces the shaft to a certain deformation which remains constant. The shaft experiences alternating stress as it turns, but the shaft usually does not vibrate.

The term "whirling" refers to the shaft vibrating while it is rotating. A shaft may whirl in the direction of rotation (forward whirl) or opposite to the direction of rotation (backward whirl). An Nth order whirl implies N oscillations for each rotation. First order whirl, the most severe, is called "whipping".

Lateral shaft resonances are usually above the operating speed range as far as shaft rotational frequency is concerned and, in the low power

range, as far as blade frequency is concerned. Excessive lateral vibration is usually a result of mechanical damage rather than poor design. The logical things to check for damage are the propeller, bent shaft, bearing damage, and alignment of flanges and couplings.

The fundamental frequency can be approximated by any one of several formulas. The following is "Jasper's Formula," which comes from Reference 6-2.

$$f_n = \frac{1}{\pi} \sqrt{\frac{3EI}{L_o^2 (4 L_o + 3 L_s) (m_p + m_{es})}}$$

where m_p = Propeller mass + 10 percent entrained water

m_{es} = .38 x shaft mass

L_o = Distance from propeller c.g. to center of propeller bearing, in.

L_s = Distance from center of propeller bearing to nearest line bearing, in.

6.5.4 Turbine Vibration

Steam turbines and gas turbines are carefully balanced as they are manufactured or installed and there are generally no significant excitations associated with them unless there is mechanical damage or the excitation comes from outside the turbines themselves.

Propeller excited vibration may be present in the turbines in the axial or torsional direction, transmitted through the main shaft, gears, and couplings. Excessive axial vibration may cause damage to the turbine thrust bearing. Excessive torsional vibration is likely to cause a failure outside of the turbine itself, such as in the couplings, gears, or shafts.

Vibration at other frequencies; such as one or two times rotational, can be due to a number of causes, and the reader is referred to the troubleshooting chart in Section 6.4.1.

The lateral vibration of turbines is often continuously monitored and, when the amplitudes exceed limits set by manufacturers, the turbine is shut down or scheduled for maintenance.

Another type of vibration problem sometimes found in turbines is the vibration of the turbine blades. It may be a single blade or, if the blades are shrouded, a section of blades. Sometimes the blades fail from fatigue. This type of vibration is not apparent from simple vibration measurements, and a vibration engineer should be consulted for this type of failure.

SECTION 7.0

EXAMPLES OF SHIPBOARD VIBRATION PROBLEMS

7.1 INTRODUCTION

The vast majority of all shipboard vibration problems the reader may encounter have occurred at other times on other ships. The objective of this section of the Guide is to give examples of these commonly occurring ship vibration problems. Through these examples it is hoped that the reader may gain some practical background and insight into the subject of ship vibration in general, and also the methods and techniques used to analyze and solve ship vibration problems.

The examples contained in this section are arranged by ship element as described previously, i.e., hull girder, major substructure, local structure, main propulsion machinery, and equipment. Each example will include a brief description of the problem, commonly observed symptoms, a discussion of the cause(s), and potential solutions. Many of the examples may be applied directly to solve identical or similar problems encountered by the reader. Others will provide the reader with a feeling for the nature of ship vibration, interpretation of observed symptoms, and the various methods used to find solutions. With the information contained elsewhere in this Guide and in the examples the reader, confronted with a vibration problem, should be able to determine the cause and potential solution for relatively simple or routine cases and may be able to implement a solution with equipment and material onboard the ship. In the case of more complex or nontypical vibration problems, this Guide provides sufficient information for the reader to approach the problem intelligently through the discovery of the symptoms and possible causes, and to plan a suitable course of action to be taken to solve the problem. Such a plan of action may include the application of a simple solution, determining where, how, and when vibration measurements should be taken or, at least, acknowledgement that an outside consultant is required, whether he be an equipment manufacturer's representative or a vibration consultant.

Before getting into the examples it is useful to look at some statistics on ship vibration problems in order to determine what the most common problems are and into what ship element types they fall. Table 7-1, derived from Reference 7-1, shows a breakdown of where vibration problems occurred on 70 ships reporting problems. As can be seen from Table 7-1, the most frequently reported problems are associated with local and major structures.

Table 7-1. Problem Locations

| SHIP ELEMENT TYPE | LOCATION | NUMBER OF SHIPS | TOTAL |
|--------------------------------------|-------------------------|--------------------|-------|
| <u>Hull Girder</u> | | | 2 |
| <u>Major and Local Structure</u> | Superstructure | 34 | |
| | Cracked Aft Peak Frames | 5 | 39 |
| <u>Main Propulsion Machinery</u> | Main Engine (Diesel) | 8 | |
| | Main Shaft Vibration | 8 | |
| | Main Shaft Bearings | 3 | 19 |
| <u>Equipment</u> | Boiler Tubes | 1 | |
| | Auxiliary Engines | 6 | |
| | Auxiliary Machinery | 3 | 10 |
| TOTAL NO. OF SHIPS | | | 70 |

Of these cases, the greatest number are in the superstructure (or accommodation spaces). This might be expected as it is where the crew spends most of their time and, hence, they are more sensitive to and critical of vibration in this area. Superstructure vibration problems can be caused by excessive global hull girder vibration, superstructure resonance or near resonance due to insufficient stiffness, or local resonances. These local resonances may be in the structure such as decks or bulkheads, but they may also be in equipment installed in the superstructure, such as joiner panels and doors, ventilation ducting, pipes, lighting fixtures, furniture, navigation equipment, etc. In light of the latter case, the number of ships with structural problems in the superstructure is probably overstated in the table, while the number with equipment problems is understated. In either case, the table does point out the very high susceptibility of the living accommodations to vibration problems, whether they be due to structure or equipment.

The second most frequently reported problems are those associated with the main propulsion system. While excessive vibration in the superstructure is very annoying, excessive vibration in the propulsion system can prove very damaging and may, in fact, limit operating speeds or render the ship inoperable.

The third most frequently reported problems, according to Table 7-1, are with equipment. If the vibration problems associated with equipment in the superstructure were included in this category then it would rival structural problems for the most frequent. Excessive vibration of equipment when it forms a part of an essential system can prevent proper operation of the ship just as with the main propulsion system.

The least frequent problems are associated with hull girder vibration. This is probably due to the ship designer's ability to accurately calculate the lower hull girder natural frequencies and to select propulsion systems which will not excite these lower hull girder modes in the operating range.

The two most frequent forms of hull girder problems are resonances of the aft end of the hull girder with blade frequency and springing. The former often manifests itself in excessive superstructure vibration at or near blade frequency. The latter is a two-noded vertical (first mode) vibration excited by wave encounter forces.

The above serves as a useful guide in determining the number and type of examples to be included for each ship element type. Major emphasis will therefore be placed on local and major substructure, main propulsion machinery, and active and passive equipment. Hull girder problems will receive the least amount of attention in the examples.

7.2 HULL GIRDER VIBRATIONS

Example 7.2.1 - Hull Girder Springing

A 1,082-foot long oil tanker on trials was ballasted down to approximately two-thirds of its full load displacement in order to conduct a 4-hour economy run. Once on course and making about 16 knots, the hull began to flex or spring in its two-noded vertical mode. Seas were approaching from about 10 to 15 degrees off the bow and were averaging approximately 40 feet between crests.

Symptoms

The hull girder was experiencing visible two-noded vertical vibration with an estimated amplitude of 12 to 18 inches. This motion was especially visible when standing on the main deck aft and slightly forward along the deck edge. The frequency was low enough to be counted and was about 32 cycles per minute. The springing was very noticeable in the pilothouse seven levels above the main deck where the vertical motion of the stern was translated into a fore and aft motion. The springing was also very noticeable in the lower engine room as a vertical heaving motion.

Problem

Springing occurs when the range of frequencies of the wave energy includes the two-noded vertical natural frequency of the hull girder. The waves, even though they may not be very large, act as a somewhat random excitation at the bow of the ship causing the hull girder to vibrate. This phenomenon most often occurs in large ships with low resonant frequencies, such as Great Lakes bulk carriers. However, occasionally it occurs in large oceangoing vessels as in this example. Figure 7-1 shows the hull girder mode shape and vibratory response at the stern of the tanker in this example. For fast ships with relatively light scantlings, hull girder stresses due to springing can be of the same magnitude as those caused by wave bending and thus may be of concern from

the point of view of structural fatigue and buckling. Springing is a function of ship speed, the heading, sea conditions, bow and stern lines, ship length, distribution of cargo or ballast weight, and the amount of energy contained in the sea. All ships experience springing to some extent. However, it only becomes a concern when the frequency of the first hull mode is within the range of frequencies of the wave spectrum. This occurs primarily with ships over 900 feet in length. Some recent experience indicates the existence of a similar phenomenon occurring in ships which are very flexible in torsion, such as container ships. In this case, torsional resonance of the hull girder may be excited in oblique waves.

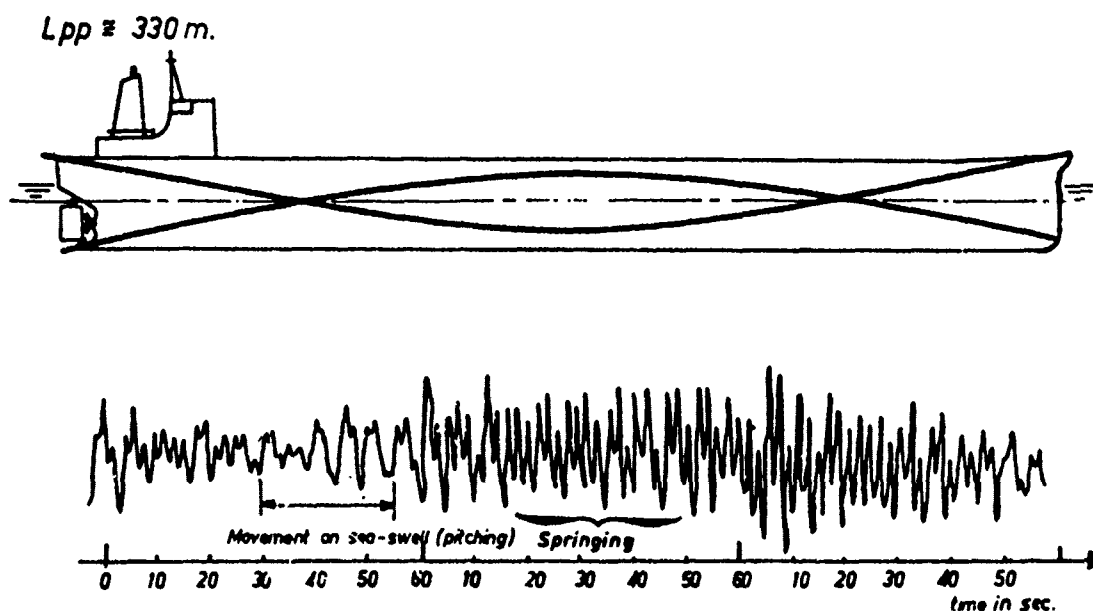


Figure 7-1. Springing Phenomena

Solution

Springing response can be reduced by changing the frequency of encounter with the waves by changing speed or heading. Usually, significant changes in these parameters are required in order to reduce springing. Another potential solution is to change the ship's distribution of cargo or ballast in order to change the hull girder natural frequency, although very large changes are required to gain only small reductions in springing. This is usually not feasible once at sea, and may not be feasible at all for most ships.

7.3 MAJOR SUBSTRUCTURE VIBRATION

Example 7.3.1 - Propeller Cavitation Induced Stern Vibration and Noise

The crew of a new 720-foot container vessel complained of excessive noise in their quarters. The crew's quarters were aft. The ship was propelled by a 32,000 SHP steam turbine turning a 6 bladed propeller at 110 rpm.

Symptoms

A noise and vibration survey was conducted on the vessel during a full load transatlantic voyage. An Askania vibrograph equipped with a 5:1 magnification probe was used to measure vibration levels in the steering gear room, engine room and in the aft deckhouse. A sound level meter was used to measure noise levels in these same areas. The results of the survey were as follows:

- a. The noise which was causing crew complaints was in the after section of the ship. The noise consisted of crackling and occasional violent "hammer-like" blows on the hull in the vicinity of the propeller. Noise level readings in the quarters generally showed dBA readings above the comfort range.
- b. Blade frequency vibratory displacement amplitudes were found to be moderate at full power and there was no indication of any resonances in the upper speed range.
- c. Vibration measurements showed strong displacements at multiples (2,3,4,5, and 6) of blade frequency.

Problem

The crackling noise and "hammer-like" blows detected in the vicinity of the propeller are generally associated with propeller cavitation. Likewise, the strong vibratory displacements at multiples of blade frequency are known to be generally caused by propeller cavitation. While the vibration levels in the crew living spaces at multiples of blade frequency did not fall in the uncomfortable range, the noise created by the vibrating decks and bulkheads created noise levels in excess of recommended comfort levels.

Based on these findings, a cavitation study was performed in a model basin to determine the amount and cause of the cavitation. The amount of cavitation was found to be excessive when a propeller blade passed the vertical or 12 o'clock position. This condition was caused by very poor water inflow to the upper area of the propeller disc.

Solution

The model basin recommended that flow directing fins forward and above the propeller be considered as a solution. Several configurations were tested in the tank and one selected which best improved the flow into the top of the propeller disc, thereby reducing propeller cavitation.

These fins were installed on the vessel and a new vibration and noise survey conducted with the following results:

- a. Noise measurements in the crew quarters showed a general 2 to 7 dBA decrease. Furthermore, the

crackling and "hammer-like" blows were no longer evident in the vicinity of the propeller.

- b. Blade frequency vibration displacements which were never considered excessive remained about the same.
- c. The amplitudes of multiples of blade frequency (2,3,4, and 5) were generally lower in the vicinity of the propeller and most importantly in the living spaces.

The installation of the fins improved flow to the propeller, reduced blade cavitation and its characteristic crackling and hammering noise, decreased the vibratory displacements at multiples of blade frequency and reduced noise levels in the living quarters resulting in improved habitability.

7.4 LOCAL STRUCTURE VIBRATION

Example 7.4.1 - Excitation of Mast Due to Vortex Shedding

A commercial cargo vessel had as part of its cargo handling equipment two masts, one on either side of a small cargo hold. These free-standing masts each supported a 5-ton boom. After being installed it was noticed that occasionally the masts would vibrate visually, up to 6 inches at the top. Since no machinery on the ship was yet in operation, the only other possible source of excitation was the wind.

Symptoms

An Askania vibrograph was used to determine the vibration frequency by holding its probe to the mast about 6 feet above the deck to which it was attached. The paper tape was analyzed to determine the frequency of vibration, which was about 5.0 Hz. A hand-held wind speed indicator was used to determine the wind speed at which the masts began to vibrate which was about 31 miles per hour.

Problem

The natural frequency of the cantilevered mast was computed using its structural parameters. The mast was 30 feet high and was made from a 24-inch diameter steel pipe with a wall thickness of 1/2 inch. Using the formula for a cantilever beam

$$f_n = \frac{a_1}{2\pi} \sqrt{\frac{EI}{\mu_1 l^4}} \text{ Hz}$$

where: $a_1 = 3.52$ for the fundamental mode

$$E = 30 \times 10^6 \text{ psi}$$

$$I \cong R^3 I = 2714 \text{ in}^4$$

$$\mu_1 = \text{Mass per unit length} = .0277 \text{ lbs/in}$$

$$L = \text{Length} = 30' = 360 \text{ inches}$$

the natural frequency was determined to be 4.7 Hz. The Kármán vortex shedding frequency has a definite relationship to wind speed, V, and cylinder diameter D, expressed by the formula:

$$f = 0.22 \frac{V}{D} \text{ Hz}$$

where f is the frequency and the constant 0.22 is known as the Strouhal Number. The vortex shedding frequency is, therefore,

$$f = 0.22 \frac{31 \frac{\text{MILES}}{\text{HR}} \times \frac{5280 \text{ FT}}{\text{MILE}} \times \frac{\text{HR}}{3600 \text{ SEC}}}{2.0 \text{ FT}} = 5.0 \text{ Hz}$$

which is very close to the resonant frequency of the mast. The mast then is in resonance with the wind generated vortex shedding frequency.

Solution

Since a 31 MPH wind speed would be experienced frequently during ship operation a solution had to be found. Several options were available:

1. Stiffen the mast structure to raise its natural frequency.
2. Stiffen the mast by adding stays
3. Add vortex spoilers to the mast to reduce the shedding forces.

Option 1 was thrown out since a practical increase in stiffness would not raise the wind speed at which resonance would occur enough to avoid the problem. Option 3 is often used on land based smoke stacks and would consist of installing small tabs arranged in a spiral around the mast or as a perforated shroud around the upper part of the stack. Both of these modify or spoil the vortex shedding phenomenon, thereby, decreasing the exciting forces. Option 2, the addition of stays, was not only the easiest solution to try, but also was the simplest and least expensive to install. For these reasons the shipyard decided to try the stay fix. Three 3/4-inch diameter wire rope stays with turnbuckles were attached to each mast as well as a jumper stay between the two masts. After the installation of the stays, there were no more reports of them vibrating. On trial the masts were watched carefully, but no vibration was detected over a range of relative wind speeds of 0 to 63 miles per hour. No reports of

the masts vibrating were ever received from the owner so it was assumed that the addition of the stays was a satisfactory solution.

Example 7.4.2 - Rudder Vibration

A commercial container vessel experienced serious cracking of its rudder horn and rudder after only 2-1/2 years of service. The ship had a 32,000 SHP steam turbine propulsion plant driving a 4 bladed propeller at 112 rpm. The stern arrangement is shown in Figure 7-2, along with the locations of cracks in the rudder and rudder horn. Vibration was suspected as the cause since a review of the design calculations showed no lack of strength.

While the above vessel was awaiting drydocking and repair, a newer sister ship which had been in service a little less than 2 years was surveyed during a full-load run. Transducers were installed down in the bottom rudder horn, on the steering gear flat (on centerline just forward of the rudder stock), and on the main deck at the centerline and port and starboard shell. Figure 7-3 shows a transverse section through the stern in way of the rudder horn and the transducer locations. At location One, in the bottom of the horn, two transducers were arranged to pick up transverse and longitudinal vibration. At all other locations two transducers were arranged to pick up vertical and transverse vibration. Due to equipment limitations, only one transducer could be monitored at a time.

During a gradual build-up of power, the two transducers in the bottom of the horn were monitored with the results shown in Figure 7-4. It is obvious from this figure that the transverse vibration of the horn was quite large and that, near full power, was close to or at resonance with blade frequency of $(111 \times 4)/60 = 7.4$ Hz. Longitudinal vibration did not appear to be a problem. With the vessel operating at full power, approximately 111 rpm, measurement at all locations were recorded and are shown superimposed on Figure 7-3 as mils double amplitude.

Problem

The most obvious thing about these measurements is the large amplitude of the horn relative to the rest of the stern structure. This fact immediately leads to the conclusion that the rudder/rudder horn combination is in resonance with the blade frequency at full power and is probably causing the cracks due to metal fatigue. Since phase relationships between the various measurements were not known, the exact mode shape of the horn and stern section could not be determined. However, from the measurement shown in Figure 7-3, it does appear that the vibrating rudder/horn is forcing the entire stern structure to rotate somewhat about its longitudinal centerline. Figure 7-5 shows the most probable mode shape of the horn and stern section.

Solution

Since a rudder/rudder horn resonance is the most probable cause of the cracking problem, what must be done is either change the exciting frequency or change the natural frequency of the horn. The exciting frequency could be

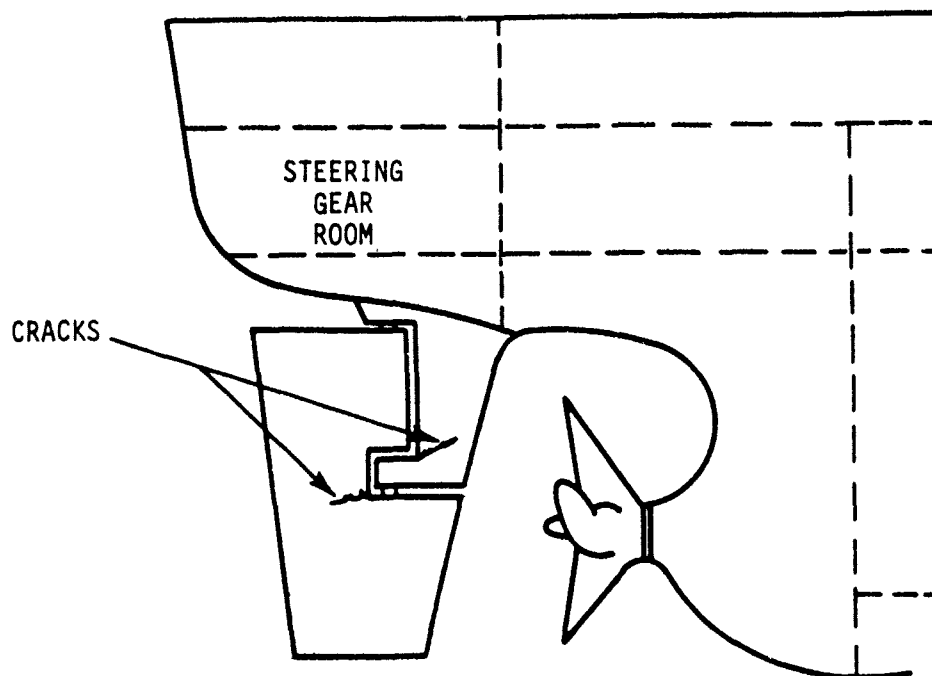


Figure 7-2. Stern Arrangement

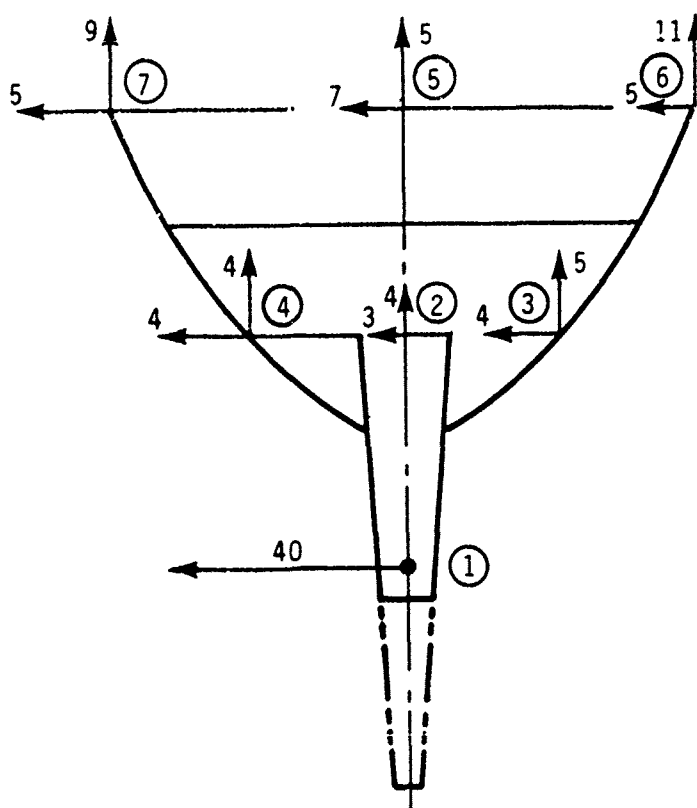


Figure 7-3. Section Through Stern

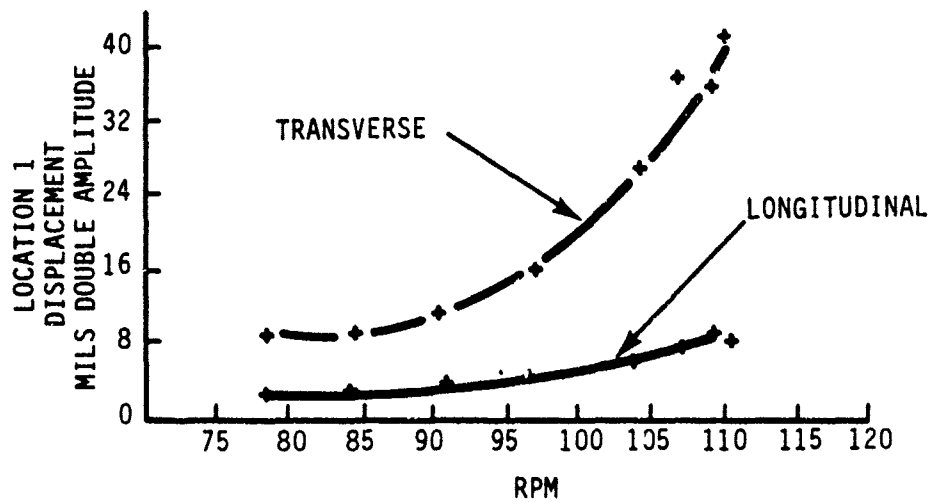


Figure 7-4. RPM vs Vibration Level of Horn

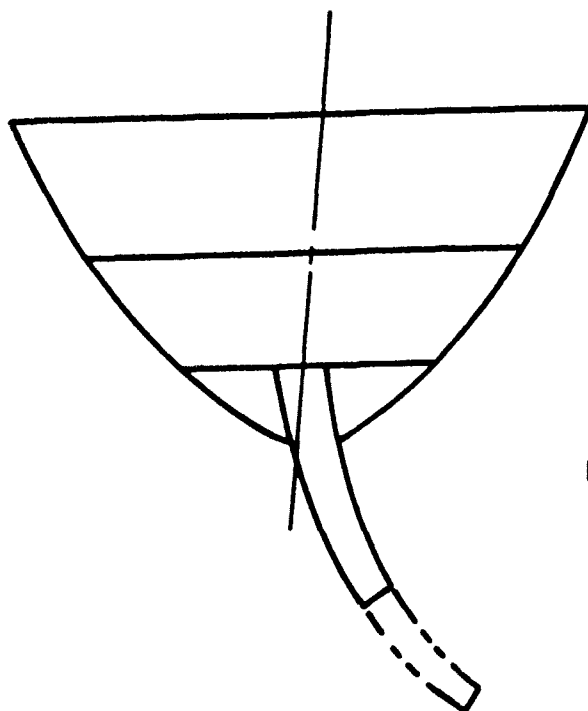


Figure 7-5. Horn and Stern Mode Shape

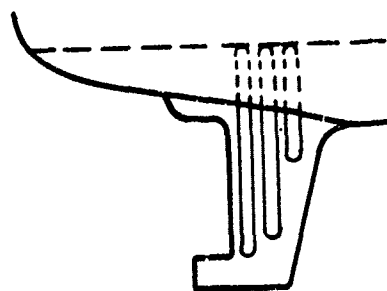


Figure 7-6. Rudder Horn Doubler

changed by fitting a new propeller with a different number of blades. Since resonance appears to be at or above 111 rpm (7.4 Hz) from Figure 7-4 an increase in the number of blades to 5 would cause the rudder horn to resonate around 89 rpm and 6 blades around 74 rpm. This solution was not acceptable to the ship owner due to the cost of new propellers and potential problems with the propulsion system. A reduction in blade number to three would put the horn resonance well above the operating speed range (149 rpm) but was not considered feasible.

The only possible solution remaining was to change the natural frequency of the rudder horn. It was proposed to increase the horn inertia by 40 percent by the addition of three vertical doublers to each side of the horn. Calculations had shown that this would increase the natural frequency about 15 percent to 8.5 Hz or 128 rpm.

While the cracks in the rudder and horn were being repaired the doublers were added as shown in Figure 7-6. In addition, great care was taken in the crack areas to reduce stress concentrations by grinding welds smooth and in-place stress relieving.

A transducer was placed in the rudder horn once the repair was completed and measurements taken during a gradual buildup in speed. The ship had no cargo. However, it was ballasted down so that the after draft was about the same as the full load draft. The transverse vibration of the horn is shown in Figure 7-7 along with the original measurements taken on the sister ship. Vibration levels of the modified rudder horn at 111 rpm were reduced to less than one half those of the original, and it appeared that the rudder/horn natural frequency had been raised well above the blade exciting frequency.

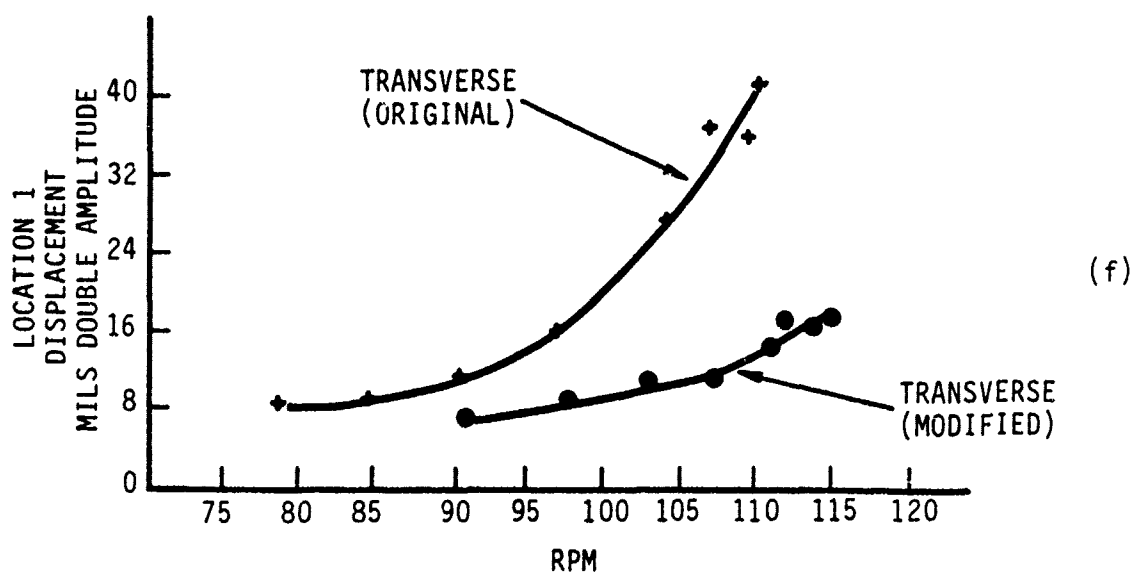


Figure 7-7. RPM vs Vibration Level of Horn

The sister ship was drydocked for a similar repair several months later. Cracks had been detected by the ships crew who had been alerted to the problem. As far as is known no further problems have occurred on these vessels.

Example 7.4.3 - Local Structural Resonance

A RoRo vessel, shown in Figure 7-8, experienced severe vibration of the main deck aft of the house on trials. The ship was almost identical to others built by the same yard which experienced no such problems. The main difference was that this ship had an 8-foot greater beam. the problem was discovered to be most severe at full power or 112 rpm. The ship had a 6-bladed propeller.

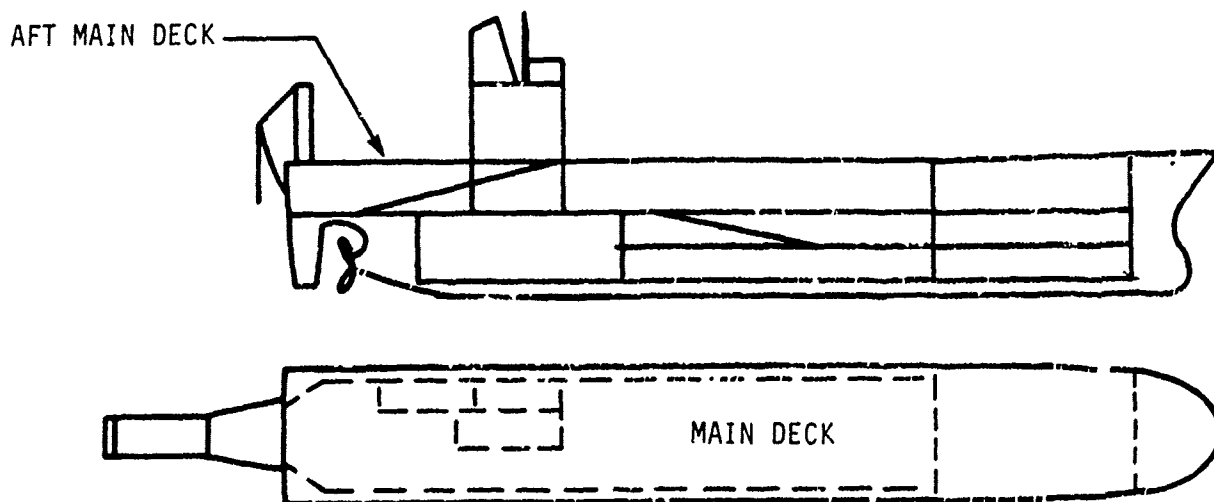


Figure 7-8. RoRo Vessel

Symptoms

At full power the main deck aft vibrated so severely that standing in the middle of the deck caused a persons teeth to chatter. The vibration level became much less severe toward the sides of the ship, at the transom, and forward toward the house. In fact, vibration levels in these locations and over the rest of the stern felt to be quite acceptable.

A vibration meter was used to measure the vibration level in the center of the deck and directly below in the center of the second deck during an incremental buildup to full power. Figure 7-9 shows a plot of the measured vibration level in mils double amplitude (d.a.) at these two locations vs. rpm.

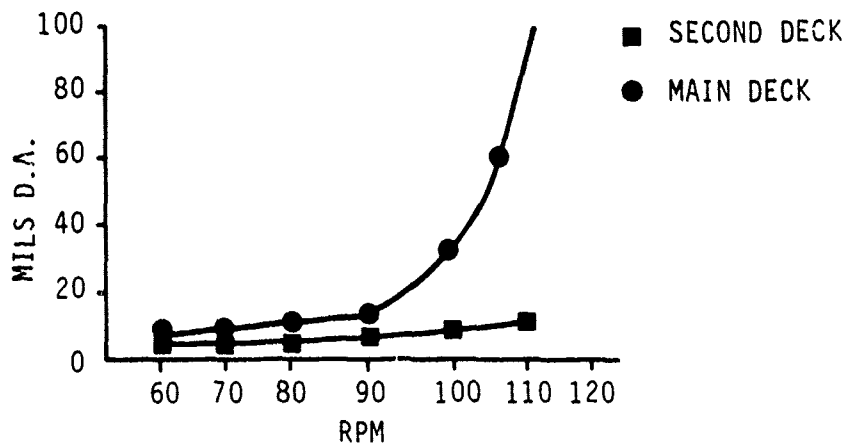


Figure 7-9. RPM vs. Amplitude

Problem

Referring to Figure 7-9, look first at the second deck. The curve shows no significant vibration at all, and even at full power at 112 rpm is only 12 mils d.a. On the SNAME T&R Bulletin 2-25 vibration chart the deck falls only slightly above the "acceptable" range for crew. This is actually quite good for the stern of a 30,000 SHP directly above the propeller.

The curve for the main deck vibration tells a quite different story. The vibration level at full power is over 100 mils d.a. and is far above the acceptable range. The curve also shows that at low rpm no real problem exists. However, beginning about 90 rpm, the vibration level increases rapidly. Such behavior leads one to suspect a resonance of the main deck with propeller blade frequency. Further evidence that the problem is a resonance is that only the center portion of the main deck experienced serious vibration, as shown in Figure 7-10. Since the curve for the main deck is still increasing at full power, actual deck resonance may be slightly above 112 rpm, or $112 \times 6 \times 1/60 = 11.2$ Hz. Why hadn't this problem occurred on the previous ships? The only difference was greater beam. Greater beam in this case meant a longer span for the transverse web frames supporting the main deck, resulting in a lower natural frequency. Unfortunately, the 8-foot increase was enough to bring the deck natural frequency down to a near resonance with blade frequency.

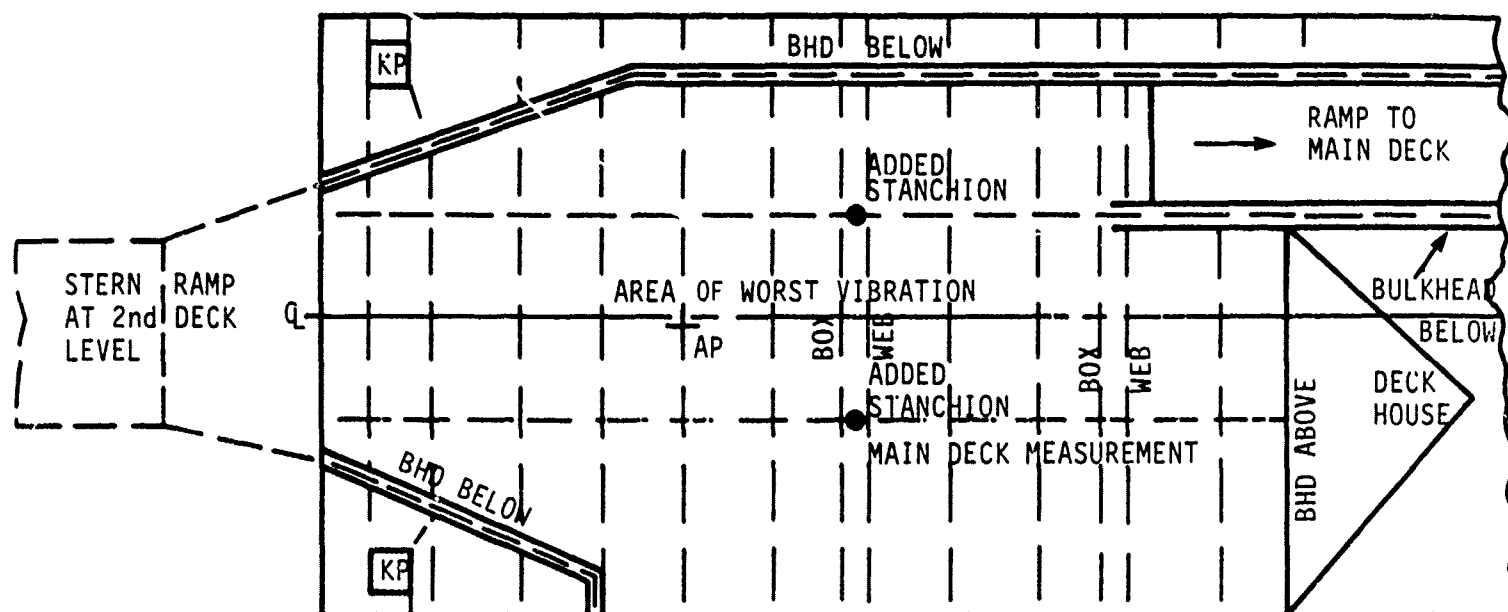


Figure 7-10. Aft Main Deck Structure

Solution

The most obvious solution would be to raise the deck natural frequency by decreasing the span of the transverse webs to what it was on the previous ships. This could be easily accomplished by using stanchions or deeper side webs with larger brackets. This was not an acceptable solution, however, since they would interfere with the flow and stowage of the RoRo cargo on the second deck.

Another way to increase the deck natural frequency would be to increase the stiffness of the transverse webs. This could be accomplished by making them deeper and increasing their flange areas. However, there was only 12 inches between the top of the RoRo cargo and the bottom of the existing webs.

Another potential solution is to lower the deck's natural frequency so that it falls below the operating range. This could be done by decreasing the stiffness of the transverse webs or by adding mass to the deck. Decreasing stiffness was not possible because of the resulting loss of deck strength. The addition of mass to the deck actually occurs when it is loaded with cargo and the natural frequency is decreased to about 7.0 Hz or 70 rpm. In this condition vibration levels would be acceptable.

The resonance, then, is only a problem when the deck is not loaded with cargo. The solution proposed by the shipyard and a vibration consultant was to add several large stanchions between the main and second deck, as shown in Figure 7-3. To overcome the objection of the owner concerning cargo flow and stowage, these stanchions were made portable. They were hinged at their attachment to the main deck transverse box webs and bolted to the second deck. A

tackle was provided to swing the stanchions up and out of the way so that they stood completely above the lower flanges of the transverses. When the ship was operated with no cargo on the main deck the stanchions were lowered and bolted into position. This operation took three seamen less than 1/2 hour to accomplish.

The resulting vibration level at the center of the main deck at full power was thereby reduced to approximately 25 mils d.a. which, although excessive for human occupation, was considered acceptable for a cargo area.

7.5 MAIN PROPULSION SYSTEM EXAMPLES

Problems encountered on ships' propulsion systems are usually excited by the propeller or the engine. The vibration may be evident in the shafting system or one of the propulsion system components. The following examples are chosen in part to illustrate how the components are interdependent.

Example 7.5.1 - Longitudinal Condenser Resonance

A commercial tanker had a conventional steam turbine/reduction gear plant as shown in Figure 7-11. At about 85 percent of full power the fore-aft motion of the condenser was visually apparent and some of the steam pipes attached to the low pressure turbine were vibrating up to 1/4 inch (d.a.).

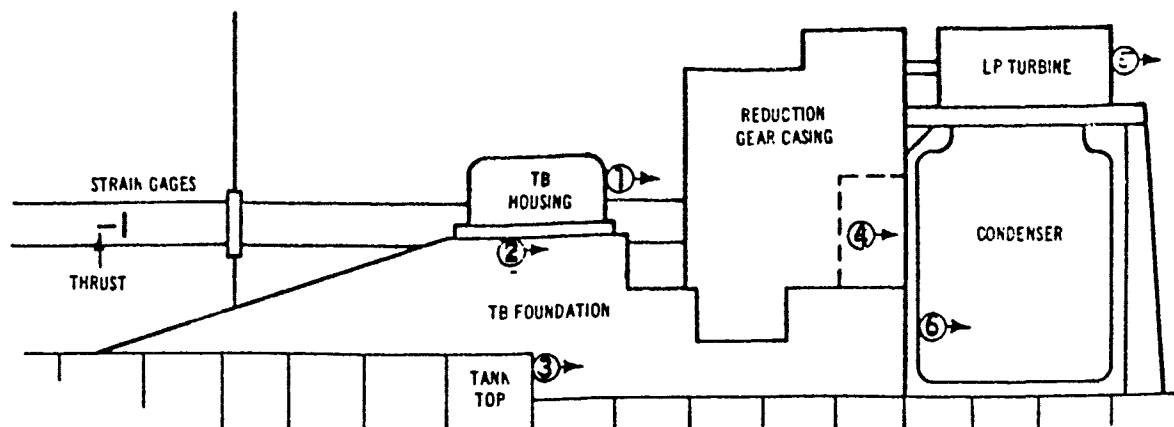


Figure 7-11. Machinery Arrangement of Commercial Tanker with Condenser Vibration

Symptoms

Vibration measurements were made in the longitudinal direction at several locations. Also, the alternating thrust in the propeller shaft just aft

of the thrust bearing was measured. The data was analyzed with a spectral analyzer. At 93 rpm all the measurements peaked at the average vibration levels shown:

| <u>LOCATION</u> | <u>AMPL AT 93 RPM</u> |
|---|-----------------------|
| (1) Top of Thrust Bearing Housing | 4.0 mils |
| (2) Top of Thrust Bearing Foundation | 3.5 mils |
| (3) On Inner Bottom below Thrust Bearing Foundation | 3.5 mils |
| (4) Forward end of Reduction Gear Casing | 4.0 mils |
| (5) Forward end of Low Pressure Turbine | 9.0 mils |
| (6) Bottom of Main Condenser | 28.0 mils |

As is usually the case, the amplitudes modulated, and during turns the vibration was considerably higher. For example, the maximum repetitive value for the condenser during a turn was 86 mils s.a. Throughout the operating speed range, the blade frequency component was predominant. The alternating thrust in the shaft did not peak at 93 rpm, but varied approximately as the square of the rpm, as would be expected if there were no effects of resonance. At full power the average blade frequency alternating thrust was about 18,000 pounds, s.a., or about 5 percent of the mean thrust.

Problem

The longitudinal natural frequency of the propulsion system was calculated by conventional methods, and was found to be greater than 11 Hz, while the 93 rpm resonance corresponds to 6.2 Hz. This fact, along with two other factors, indicate that this is not the normal longitudinal resonance. First, a normal resonance involves the greatest amplitudes at the propeller, and the amplitudes get smaller on the inboard components. The thrust bearing would probably have the greatest motion, with the gear casing, turbines, and condenser motion being less (or about the same). On the tanker the condenser motion was seven times the thrust bearing, indicating a condenser resonance. Second, the dynamics of the shaft in a normal resonance cause the alternating thrust to peak at the resonance as does the motion of the components. The lack of a peak in that curve indicates that the propeller is not participating significantly in the resonance.

Solution

A mathematical analysis was made using a model that included a detailed representation of the condenser/low pressure turbine mass and its relatively flexible connections to the gear case and inner bottom. The calculated response was close, and by varying the stiffnesses of the connections, the natural frequency and relative amplitudes measured could be duplicated in the model. Once this model was refined, it was possible to calculate the effect of a proposed fix. Two linkages were proposed to restrain the bottom of the condenser as shown in Figure 7-12. The pinned ends would allow thermal growth in

the vertical direction. The stiffnesses were added to the model, and calculations made of the modified system for various linkage stiffnesses. For a stiffness of 1×10^6 lb/in in each linkage, the resonance occurred at 8.5 Hz, or 128 rpm, 116 percent of full power. That was considered far enough above the operating speed to be acceptable. The calculations now showed the maximum amplitudes at full power to be as follows:

| <u>LOCATION</u> | <u>MAX. AMPL</u> |
|--------------------------------------|------------------|
| Forward end of Reduction Gear Casing | 3.5 mils |
| Forward end of Low Pressure Turbine | 5.0 mils |
| Bottom of Main Condenser | 6.0 mils |

This was considered satisfactory and the stiffening linkages were recommended.

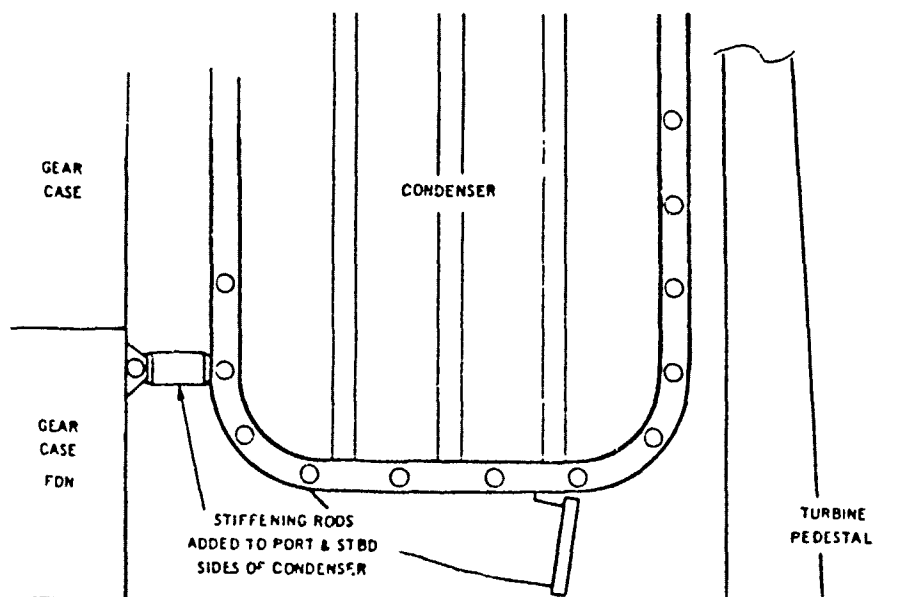


Figure 7-12. Possible Method of Stiffening Condenser Support

Example 7.5.2 - Longitudinal Resonance Just Above Full Power

A Navy helicopter assault ship had a single screw, conventional steam turbine/reduction gear plant as shown in Figure 7-13, and a four-bladed propeller. At or near full power the longitudinal vibration of the propulsion system seemed excessive and measurements were made to evaluate the system. Note

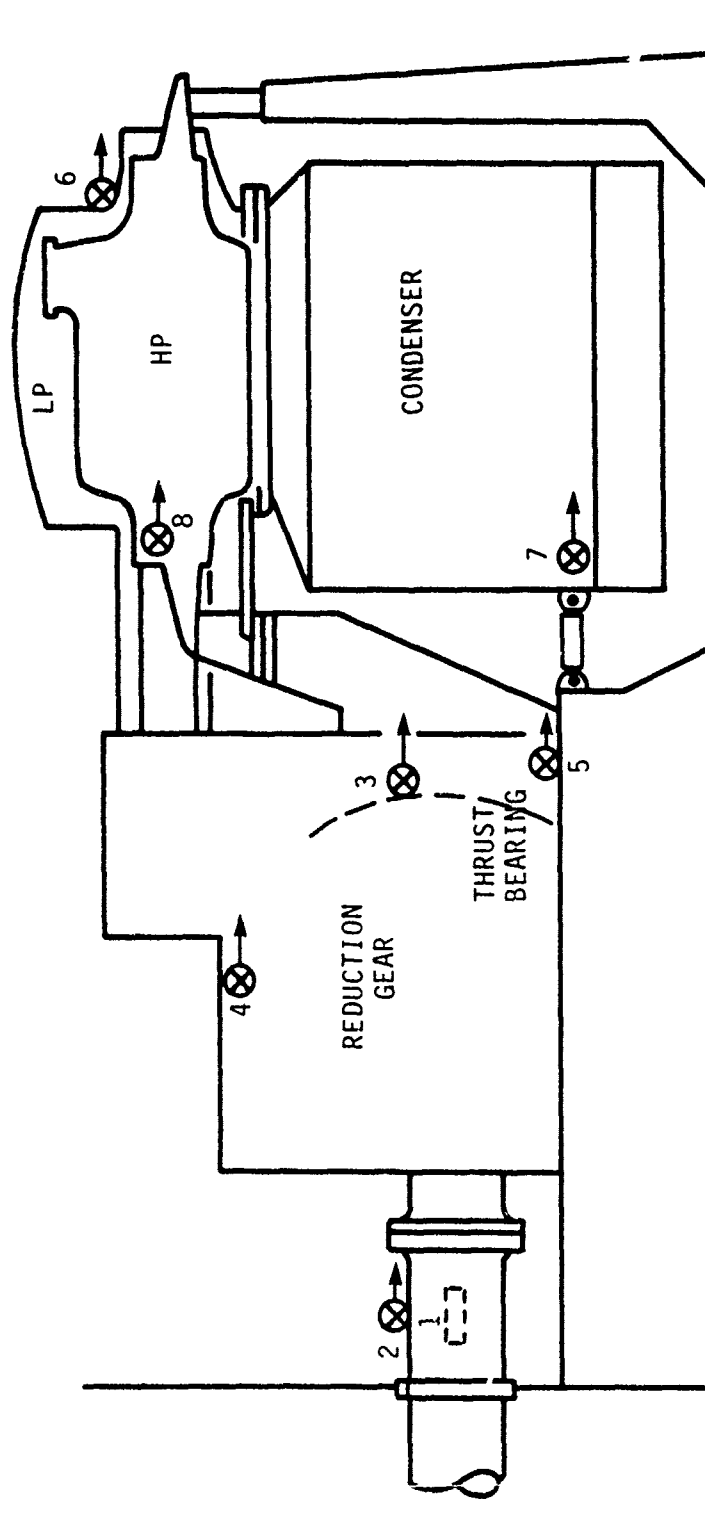


Figure 7-13. LPH Propulsion System Showing Location and Orientation of Gages

from the figure that the condenser mounting had already been modified to reduce its fore/aft vibration just as recommended for the tanker discussed above.

Symptoms

Measurements were made at various locations in the longitudinal direction on the propulsion system. In all locations the amplitudes increased significantly near full power, but there were no apparent resonances. The maximum repeatable amplitudes obtained from oscillograph records at full power were:

| | <u>LOCATION</u> | <u>AMPLITUDE</u> |
|-----|--------------------------------------|------------------|
| (1) | Alternating Thrust at Reduction Gear | 38,000 lbs |
| (2) | Shaft Motion near Reduction Gear | 11.5 mils |
| (3) | Thrust Bearing Housing Motion | 8.0 mils |
| (4) | Motion of Top of Gear Case | 10.6 mils |
| (5) | Motion of Gear Case Foundation | 5.3 mils |
| (6) | LP Turbine Casing Motion | 9.7 mils |
| (7) | Condenser Motion | 7.8 mils |
| (8) | HP Turbine Casing Motion | 12.0 mils |

During hard turns at full power the amplitudes were about twice the above amounts.

Problem

The above amplitudes were considered marginal, and were caused, in part, by the high alternating thrust (38,000 lbs amounts to about 13 percent of full power mean thrust). An even number of blades (this had four) operating behind a skeg, which is the usual single screw arrangement, often results in high propeller forces. The significant increase in amplitudes near full power indicated that a longitudinal resonance may be just above the full power rpm. To check this, double blade frequency vibration was analyzed. In all locations there was a small peak at 65 rpm, or 8.7 Hz. This would correspond to blade frequency at 130 rpm or 111 percent of full power rpm. Normally 115 percent is considered a reasonable margin.

Solution

The machinery foundation had already been stiffened and further stiffening to raise the natural frequency would involve a lot of structural work and a small change in natural frequency. A change in the number of propeller blades was considered. There were several structural resonances in the ship that would be affected by such a change, but only the longitudinal propulsion system is discussed here.

Increasing the number of propeller blades would bring the longitudinal critical within the operating range of blade frequency, but the propeller forces would be less and the end result may be an improvement. A rough

approximation of what could be expected with 5, 6, and 7 blades is shown in the following table. It is based on approximate assumed pressure force differences and an assumed magnification factor of 7 which is about average for longitudinal criticals.

| No. of Blades | 4 | 5 | 6 | 7 |
|---|------|--------|--------|--------|
| Critical RPM | 130 | 104 | 87 | 75 |
| Percent of Full Pwr RPM | 111% | 89% | 74% | 64% |
| Alt. thrust at above spds with 4 blades | ---- | 15,000 | 12,000 | 8,500 |
| Assumed % of 4-Bladed Alt. Thrust | 100% | 75% | 55% | 40% |
| Alt. Thrust at above spd with no. of bl. shown | — | 11,200 | 6,600 | 3,400 |
| Magnification Factor | 7 | 7 | 7 | 7 |
| Alt. Thrust at Resonance | — | 78,000 | 46,000 | 24,000 |

This table indicates that the number of blades would have to increase to seven in order to produce an alternating thrust at resonance which is less than that which originally occurred at full power. Although this example illustrates the considerations involved when changing the number of blades, before such a change is made, detailed calculations of the exciting forces (based on a wake survey, if possible) and the propulsion system response should be made. In the present study, after the structural problems were included in the consideration of more blades, the decision was made to keep the four bladed propeller.

Example 7.5.3 - Longitudinal Turbine Resonance

On a Naval vessel with a conventional steam turbine/reduction gear propulsion plant, the High Pressure Turbine vibrated fore/aft visibly when operating in the upper 1/4 of the rpm range.

Symptoms

Measurements were made on the HP Turbine and its foundation as shown in Figure 7-14 in the longitudinal direction. In addition, measurements were made on the thrust bearing housing and reduction gear case to get a rough evaluation of the longitudinal vibration characteristics of the entire system. The amplitudes increased with rpm. The maximum repetitive amplitudes were

obtained from oscillograph records and were the greatest at full power (144 rpm) as follows:

| <u>LOCATION</u> | <u>AMPLITUDES</u> |
|----------------------------|-------------------|
| (1) Thrust Bearing Housing | 4.7 mils |
| (2) Gear Case | 2.5 mils |
| (3) Bulkhead | 1.8 mils |
| (4) Condenser Foundation | 3.3 mils |
| (5) HP Turbine Base | 13.0 mils |
| (6) HP Turbine Casing | 17.2 mils |

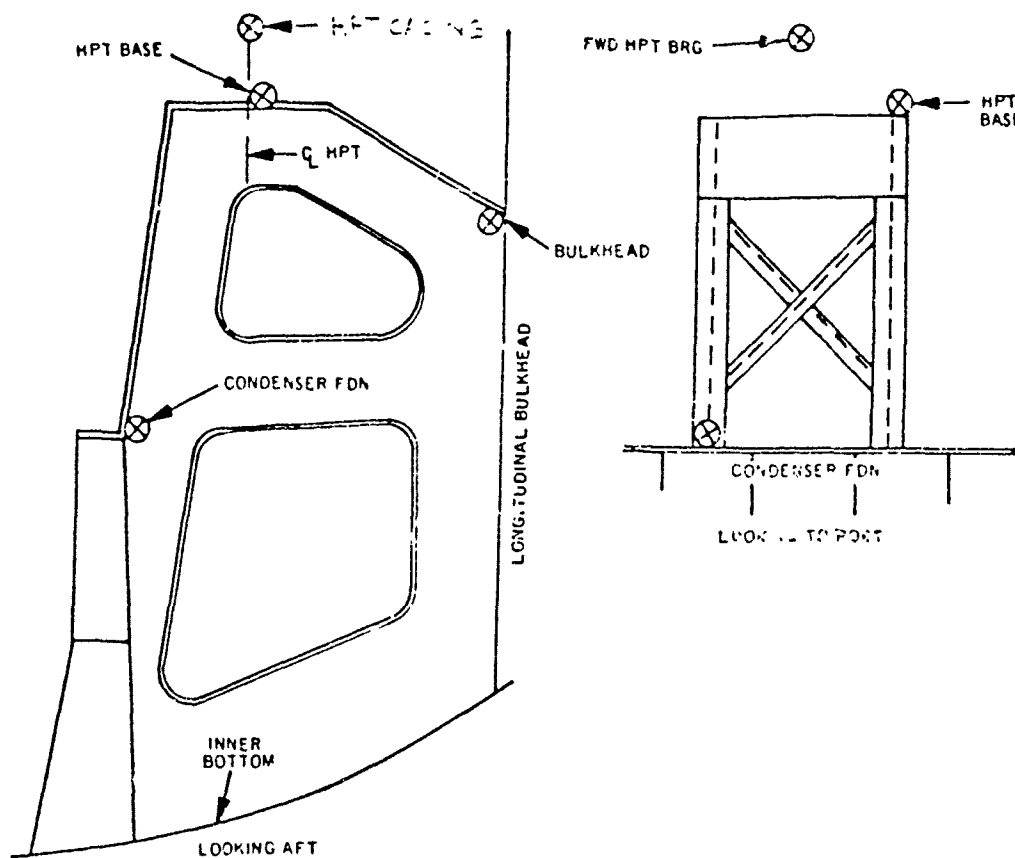


Figure 7-14. Locations of Gages on High Pressure Turbine and Foundation of AOE

Problem

The amplitudes of the reduction gear case and thrust bearing housing indicate that the longitudinal propulsion system vibration in general is not excessive. The pickups on the condenser foundation and bulkhead indicate that the structures on which the turbine foundation is built are solid enough. The high amplitudes on the turbine base and turbine casing reflect a low stiffness in the longitudinal direction of the turbine foundation, resulting in a local resonance of this unit.

The usual support for a high pressure turbine consists of a rigid connection between the reduction gear casing and the after end of the turbine, and a flex plate forward. This arrangement causes the turbine to move fore and aft with the reduction gear, keeping the vibratory displacements across the dental couplings in the gear train to a minimum. This relative motion should be minimized since the couplings are subjected to constant torque, and excessive longitudinal motion combined with the torque can cause serious coupling wear. On this ship the relative displacement between the gears and HP Turbine was as much as 14 mils s.a.

Solution

It was recommended that the stiffness of the HP Turbine foundation be increased in the longitudinal direction to reduce the relative displacement between the reduction gear and the HP turbine (across dental couplings). To do this it was suggested that:

- (1) Steel plates be welded between the flanges of the two main vertical supporting plates from the turbine base to the longitudinal bulkhead on one side and to the condenser foundation on the other side.
- (2) Triangular plates capped with flanges be welded on the HP turbine foundation on each of the four corners formed by the condenser foundation or the longitudinal bulkhead and the main vertical supporting plates of the foundation. These should intersect the vertical supporting plates as high as possible without interfering with existing deck plates and piping.

These changes are shown in Figure 7-15.

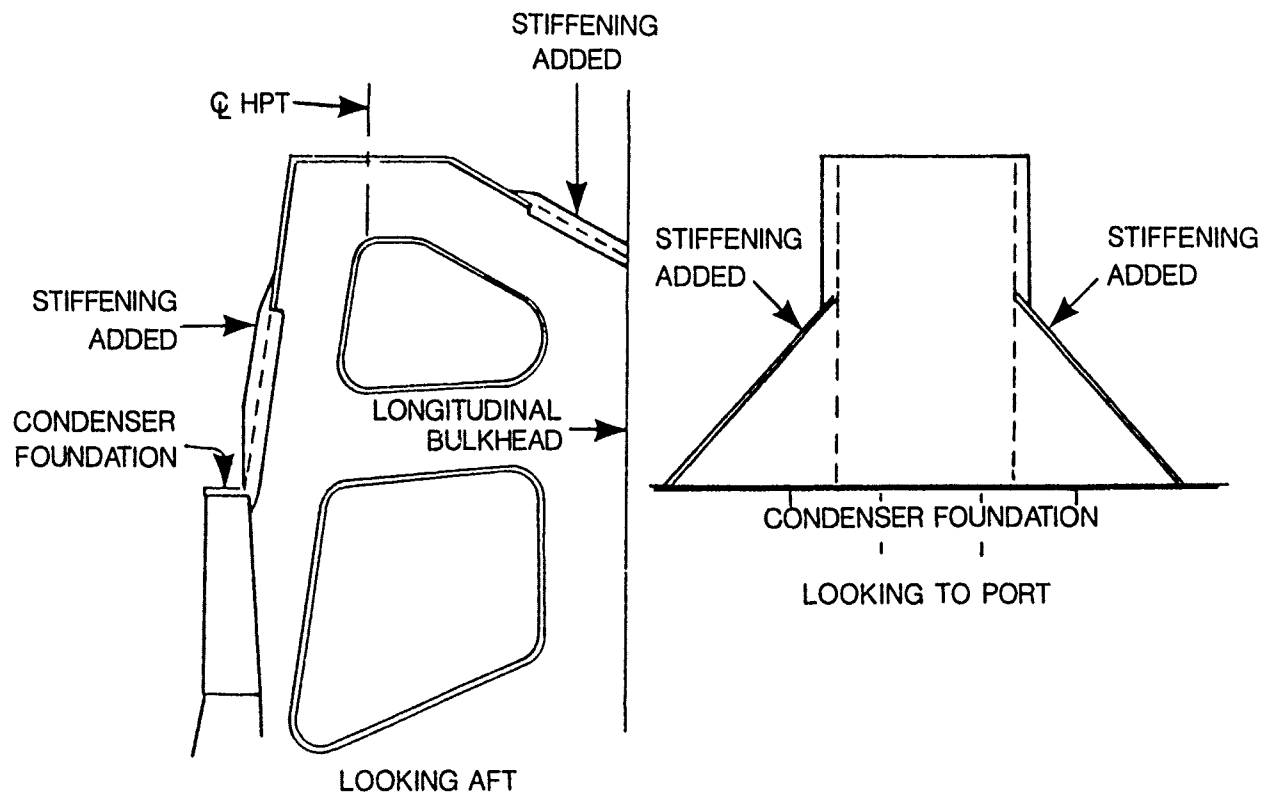


Figure 7-15. Foundation of HP Turbine with Recommended Changes

Example 7.5.4 - Coupled Longitudinal/Torsional Shaft Vibration

A river towboat with a twin-screw diesel-powered direct drive propulsion system reported excessive longitudinal shaft vibration. Some parts of the four shoe (two ahead, two astern) thrust bearings had failed along with the mounting bolts for the thrust bearings.

Problem

Design calculations indicated no longitudinal resonances in the operating speed range of blade frequency. However, a torsional resonance was calculated to be at 16.3 Hz. This could be excited by blade frequency or firing frequency in the operating speed range, so it was decided to make torsional measurements as well as longitudinal measurements. Measurements were made throughout the speed range of 100 to 275 rpm with an Askania hand-held vibrograph. There was a peak at 163 rpm in all locations, and the maximum repetitive amplitudes at that speed were:

| <u>LOCATION</u> | <u>AMPL. AT 163 RPM</u> |
|---|-------------------------|
| Torsional Displ. at Fwd end of Crankshaft | 0.25 deg. |
| Longit. Displ. of Fwd end of Crankshaft | 21 mils |
| Top of Thrust Brg Housing | 16 mils |
| Thrust Brg Housing at Shaft Level | 11 mils |
| Shaft Displ. near Thrust Brg | 21 mils |

The measurements showed all locations to be in phase, and the frequency of vibration to be 6th order. This corresponded to twice the diesel firing frequency (6 cylinder, 4 cycle engine) rather than propeller blade frequency (4 blades), which is normally the predominant component for longitudinal vibration. Since the firing frequency and its harmonics would directly excite torsional vibration, and since the frequency was close to the calculated first mode torsional frequency (163 rpm, 6th order is 16.3 Hz) it was concluded that the vibration was due to the torsional resonance. The longitudinal motion was induced in the shaft by the propeller. Due to the pitch of the propeller, the torsional motion causes an axial reaction force, and longitudinal vibration. The differences in the amplitudes at the top of the thrust bearing and at the shaft level indicate a significant rocking motion of the thrust bearing and its foundation. The differences between the shaft motion and the thrust bearing housing motion indicate that the thrust collar is bouncing back and forth from the forward to the aft shoes (the clearance was a total of 15 mils). The classical longitudinal resonance excited by propeller blade frequency forces was not detected in the operating range (up to 275 rpm). This means that the frequency was above 4×275 or 1100 CPM. The sixth order, however, could excite 1100 CPM at 183 rpm. It was believed that the 6th order excitation at 163 rpm caused a large axial response because it was a near resonant condition.

Solution

It was decided to make a change in the longitudinal characteristics of the propulsion system rather than the torsional. It was recommended that the thrust bearing foundation be stiffened in the axial direction, particularly so as to reduce the rocking motion. This would increase the longitudinal natural frequency to avoid the near resonant condition at 163 rpm where the alternating thrust is a maximum. The structural changes were made and the crew of the towboat subsequently reported that the objectionable vibration was eliminated.

Example 7.5.5 - Lateral Shaft Vibration

The builder of a diesel powered tanker complained of high levels of lateral vibration of the propeller shaft and associated equipment. The arrangement was as shown in Figure 7-16, and involved a power take-off for a generator, a line bearing, and an oil distribution box for a controllable pitch propeller, all located between the engine/thrust bearing and the stern tube bearing. The vibration was visibly apparent on all three of these components.

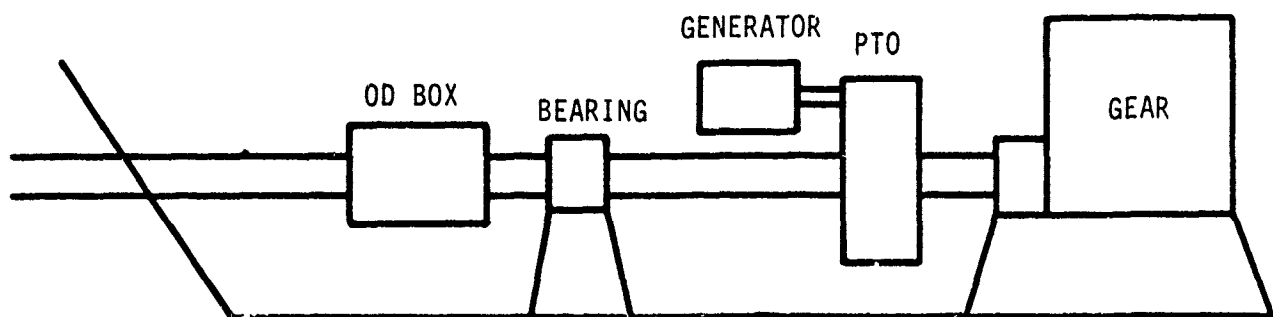


Figure 7-16. Arrangement of Tanker Propulsion System with Lateral Shaft Vibration

Symptoms

Both the O.D. Box and the P.T.O. casing were supported by the shaft with bearings, and did not rotate. Since the propeller was the controllable pitch type, the propulsion system could be operated while dockside, and measurements were made in the vertical and horizontal directions on the O.D. Box, the bearing housing, and the P.T.O. The amplitudes were higher in the horizontal direction than vertically, and greatest on the bearing and O.D. Box. The frequency of vibration was the rotational frequency and the amplitudes increased greatly near full speed.

Problem

The measurements indicated a lateral resonant or near resonant condition of the entire line shaft. The line bearing should have restrained such motion, but its own amplitudes in the horizontal direction showed that it did not. To verify the suspicions, calculations were made of the natural frequencies of the span between the stern tube bearing and the line bearing, and also of the span between the stern tube bearing and the thrust bearing/engine. The former was well above the operating range, and the latter was close to full power speed. This not only verified the problem, but also indicated that if the line bearing were stiffened the near resonant condition would be eliminated.

Solution

The foundation of the line bearing is shown in Figure 7-17, and obviously would be very flexible in the athwartships direction. A recommendation was made to stiffen this foundation as shown in Figure 7-17. The builder did so and reported the objectionable vibration had been eliminated.

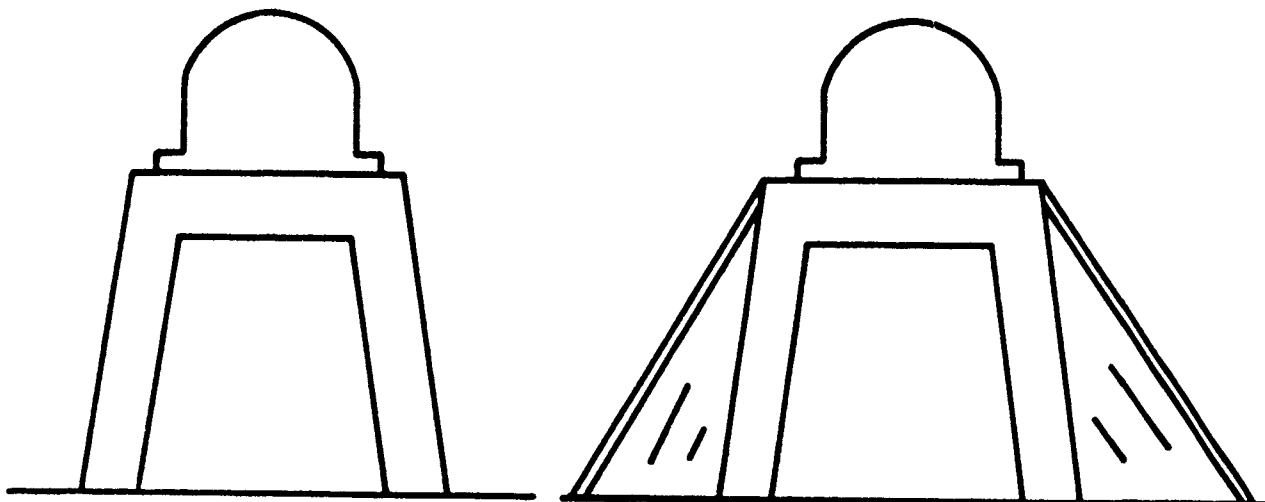


Figure 7-17. Existing and Stiffened Line Shaft Bearing Foundation

Example 7.5.6 - Torsional Shaft Vibration

A rumbling was heard in the reduction gears of a diesel powered river towboat when operating at half rpm, normal for transiting without a tow. At full speed there was no problem.

Symptoms

Torsional calculations indicated a first mode resonance near half-rpm, with the propeller and engine oscillating against each other, and a node in the line shaft near the reduction gear. Torsional displacements were measured on the propeller shaft as close to the stern tube bearing as possible in order to get away from the node. A Gelger torslograph (mechanical, belt driven seismograph) was driven by the shaft by means of a belt. The measurements showed a peak as expected, near half-speed at blade frequency. The single amplitude of the alternating torque when extrapolated by means of the calculated mode shape was equal to the mean torque, indicating that the gears were unloading on each cycle. This condition would certainly explain the reported "rumbling". It was speculated that the person who did the torsional calculations was not worried about a resonance at half speed, not knowing that much of the towboats operation would be at that speed.

Solution

In order to change the natural frequency, two approaches are obvious. First, the stiffness can be changed. In the case of first mode torsional vibration, that would mean stiffening the shaft torsionally, which involves changing shaft diameter, bearing sizes, etc. The cost would be prohibitive.

The second, is to change the mass characteristics of the system, and this is what was recommended. A 3-foot diameter disk placed as far aft as possible on the line shaft was recommended. Calculations showed that this reduced the first mode natural frequency to about 1/3 the rpm, which is below the usual operating speeds. It also introduced an additional mode at about 2/3 the rpm, but that speed was not normally used for operation, and was considered an acceptable condition. Figure 7-18 shows the propulsion plant arrangement, and the before and after mode shapes.

Example 7.5.7 - Singing Propeller

The builder of a heavy lift cargo ship with twin screws reported a loud tone permeating the after part of the ship while underway at 150 to 200 rpm. (Full power was 225 rpm.) The propulsion systems were diesel/reduction gear, and both shafts were the same length. The shaft was supported by struts outboard, with stern tubes between the hull penetrations and the struts. The strut bearings were oil lubricated.

Symptoms

The ship was asked to operate in the middle of the objectionable speed range (170 rpm), and the noise observed at various locations near the stern. The tone was mid-frequency and drowned out all other noises in the area. It was greatest near the starboard propulsion system outboard. Noise measurements showed the frequency to be 670 Hz. Vibration measurements were then made in various locations, velocity being chosen to avoid the 670 Hz signal being lost among lower frequency components. The measurements were analyzed with a 1/10 octave band General Radio analyzer, with the following results (vertical and athwartships measurements were about the same and are averaged below):

| <u>LOCATION</u> | <u>OVERALL LEVEL</u> | <u>LEVEL AT 670 HZ</u> |
|-------------------------|--------------------------|----------------------------|
| Fwd end stbd stern tube | 60 Vdb | 57 Vdb |
| Fwd end port stern tube | 56 Vdb | 42 Vdb |
| Stbd strut ends | 59 Vdb | 54 Vdb |
| Aft end reduction gear | 64 Vdb | 43 Vdb |

The next step was to measure the vibration where it was greatest (the forward end of the stern tube) for different speeds. The level was between 52 and 58 Vdb from 150 to 200 rpm, and at least 20 Vdb down outside of that range, and the frequency remained at 670 Hz independent of the rpm. At this point it was suspected that, because it was constant, the 670 Hz was a structural resonance of some outboard component of the starboard propulsion system. The excitation was present only over a certain range of rpm, and because of the broad peak, did not appear to be directly related to shaft speed. A flow excited phenomenon was suspected.

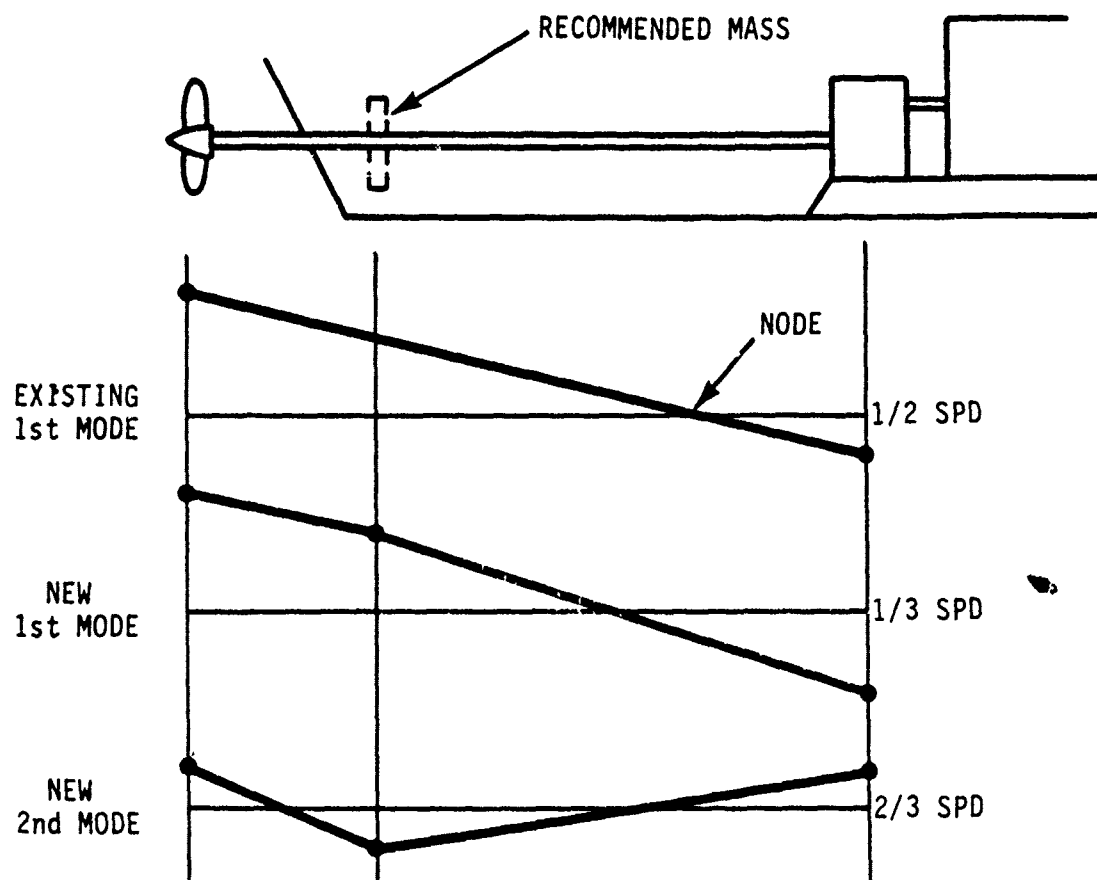


Figure 7-18. Arrangement and Mode Shapes of Towboat Propulsion System with Torsional Vibration Problem

When the ship returned to port, a diver was sent over the side with a padded sledge hammer with instructions to impact the propeller, the shaft struts, and the stern tube. At the same time the vibration of the forward end of the stern tube was measured. When the propeller was hit, the 670 Hz signal was detected, being greatest when the trailing edge was impacted. That seemed to be the object that was vibrating. The excitation must also be considered. It was suspected that Kármán vortices caused by flow past a blunt trailing edge provided the excitation. This is explained in detail in the Section on Evaluation and Correction. The frequency of the vortices can be approximated from:

$$f_s = S_n \times V/t$$

where: f_s = The shedding frequency in Hz

S_n = The Strouhal No. (about .2 according to Saunders, Reference 7-2)

V = The flow velocity in in/sec

t = The width of the trailing edge (between points of separated flow) in inches

This was used to calculate the frequencies associated with flow past the struts (less than 100 Hz) and the propeller blades (in the neighborhood of 600 - 800 Hz depending on exactly what thickness was assumed). This was considered sufficient verification to recommend a fix for vortex shedding from the trailing edge of the propeller.

Solution

Propeller blade vibration due to Kármán vortices had occurred before, and the phenomenon is called a "singing propeller". The usual fix is to change the width of the trailing edge. It could be made sharper or more blunt. On an existing propeller it is easier to make it more blunt, commonly with a chisel type edge, as shown in Figure 7-19. On the present ship the width was judged to be about 0.2 inch, and it was recommended that the trailing edge be ground down to make the width about 1/2 inch as shown. This was accomplished during the next drydocking, and the tone disappeared.

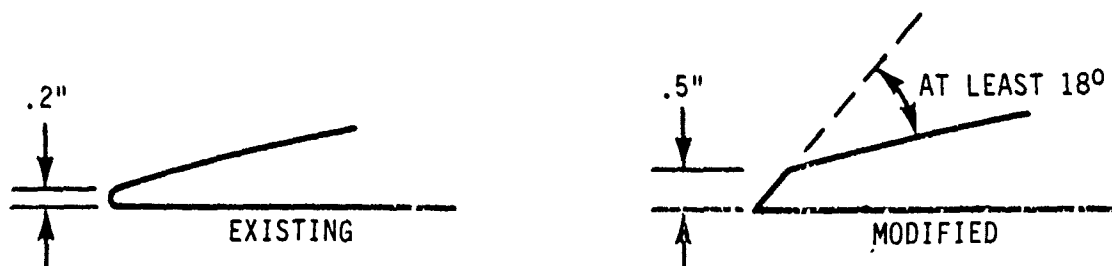


Figure 7-19. Existing and Modified Trailing Edge of Singing Propeller

7.6 EQUIPMENT VIBRATION

Example 7.6.1 - Furniture Vibration

The chief mate of a 600-foot commercial container vessel had complained without results to the ship owner about severe vibration which made it impossible to write on his desk. Finally, he sent the note reproduced in Figure 7-20 to the owner who forwarded it to the shipbuilder asking for advise.

Symptoms

The shipbuilder checked back in their records to see if any serious vibration had been reported during the trial vibration survey. One measurement in the vertical direction had been taken in the center of the floor of the Chief's office. It showed a vibration level of 2.8 mils double amplitude at 6.35 Hz which was blade frequency at full power. The engineer conducting the survey on trials had written the note contained in Figure 7-21 in the margin of his data sheet. Apparently nothing was done to fix this problem after the trial.

Problem

The 2.8 mils double amplitude at blade frequency is not severe enough to cause problems in writing. Therefore, the problem was suspected to be in the desk itself. An identical desk was found on a ship in the yard undergoing final stages of outfitting. The desk is shown in Figure 7-22. A small vibration generator was clamped to a steel deck stiffener directly below the desk to excite vertical vibration in the desk. As the generator speed was varied from 0 to 10 Hz, vibration of the desk top was recorded. As suspected, the empty desk had a vertical resonance at about 7.0 Hz. Assuming the Chief Mate had loaded his desk with the usual amount of junk the desk natural frequency may well have dropped to coincide with blade frequency. With the generator running at 7.0 Hz wooden wedges were driven between the desk and the deck, in between the legs on each side. This reduced the vibration level of the desk top to that of the deck.

SS W [REDACTED]

Chief Mate

April 27, 1973

[REDACTED]

I have complained about
vibrations of my back
to with no results. Problem
is serious as you
can see! Please do
something!

H.B.

Figure 7-20. Note from Chief Mate

| STATION NO. | TIME | LOCAL EXCITING FREQUENCY | FILTER FREQUENCY | METER READING | | | REMARKS |
|--|------|--------------------------|------------------|------------------|------------------|------------------|--------------------------|
| | | | | POSITION #1 MILS | POSITION #2 MILS | POSITION #3 MILS | |
| 47 | | 6.35 Hz (BLADE) | 6.35 Hz | 1.5 | 4.1 | 2.3 | |
| 48 | | | | 0.6 | 3.7 | 2.1 | |
| 49 | | | | 2.1 | 3.7 | 1.4 | |
| 38 | 1450 | | | 2.3 | 3.5 | 2.3 | BRIDGE DK. |
| 39 | | | | 0.9 | 4.4 | 1.5 | |
| 40 | | | | 1.6 | 4.0 | 1.1 | |
| 41 | | | | 2.3 | 3.9 | 0.6 | |
| 32 | 1500 | | | - | - | - | BOAT DK. (LOCKED) |
| 33 | | | | 1.4 | | | |
| 34 | | | | 2.8 | | | CHIEF MATE OF (SEE NOTE) |
| 35 | | | | 1.5 | | | |
| NOTE: CHIEF MATE'S OFFICE VIBRATION LEVEL ON DECK OK BUT DESK TOP VIBRATION CAUSED DIFFICULTY IN WRITING. TALK TO VENDOR. | | | | | | | |

Figure 7-21. Note Written by Engineer Conducting Survey on Trials

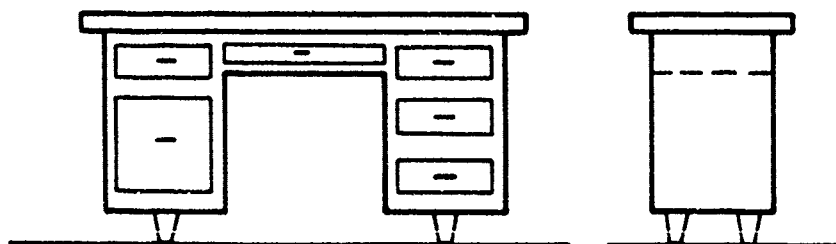


Figure 7-22. Illustration of Desk

Solution

The real problem here is that the desk had been poorly designed and lacked sufficient support structure for the legs. The desk manufacturer was notified and asked to correct the problem in all future desks.

The Chief Mate was sent a letter explaining the problem, and two varnished mahogany custom fitted wedges to drive under his desk. No more complaints were received so it was assumed the Chief was satisfied.

Example 7.6.2 - Rattle In Ventilation Duct

Two new crew members of a tanker complained that they were unable to sleep because of excessive noise in their stateroom. The source of the noise appeared to be located in the overhead above the crew's stateroom and only occurred at night.

Symptoms

The two crew members were in the habit of closing the ventilation damper in their room to reduce the temperature at night. When the damper was opened the noise stopped, but the room soon became too hot for sleeping.

Problem

After one particularly sleepless night, the two crew members removed a ceiling panel and one of them began to "feel around" trying to discover the source of the noise. When closed, the portion of the vent duct containing the butterfly damper vibrated. By placing his hands on the duct the noise level could be reduced somewhat, but not stopped. A little more "feeling around" found the area of greatest vibration. It was in the exact location of the butterfly damper valve. Further investigation found that the stem of the damper vibrated when the valve was closed, but not when it was in any other position. By applying a force perpendicular to the valve stem, the vibration and noise could be completely eliminated. Letting go of the stem caused the noise to return.

There are two factors involved with this problem. One, the butterfly damper valve stem bearings had too much clearance; and two, the shipbuilder had put a mechanical stop on the damper so that air flow could not be completely stopped. Apparently the slight amount of airflow around the butterfly damper produced sufficient forces to cause it to flutter. Over a period of years, this fluttering had caused the stem bearings to wear allowing the them to rattle which produced the annoying noise.

Solution

As is often the case on a ship, the two crew members devised a "make shift" solution. A number of strong rubberbands were looped around the valve stem and attached with a wire coat hanger to a nearby pipe. Tension in

the rubberbands was sufficient to stop the vibration of the stem and, therefore, the noise. The ceiling panel was replaced and the two crew members settled down to a good night's sleep.

A more permanent solution would have been to replace the stem bearings. However, the cost compared with the rubberbands and coat hangers would have been much greater.

7.7 REFERENCES

- 7-1. "Vibration Investigation," Helge Johannessen, Seminar of Ship Vibration, 1977, Det Norske Veritas.
- 7-2. Saunders, "Hydrodynamics in Ship Design."

Appendix A

TERMS, DEFINITIONS, VIBRATION FUNDAMENTALS

To conveniently discuss, evaluate, or communicate on the subject of vibration, it is essential that one familiarize himself with the terms and definitions commonly used in vibration theory. Thus, this section will include terms, definitions and vibration fundamentals. The notation and convenient metric conversion factors are also included, as well as a useful tabulation of natural frequencies for many common systems.

A.1 DEFINITIONS

The definitions are primarily in non-mathematical terms. For more rigorous definitions see ASA S1.1-1960, "Acoustical Terminology (Including Mechanical Shock and Vibration)," and ISO 2041, "Vibration and Shock Vocabulary."

A.1.1 Reference Terms and Definitions (Alphabetical)

| | |
|-------------------|--|
| Acceleration | Acceleration is the time rate of change of velocity. |
| Added Mass | The apparent increase in mass of a body vibrating in a liquid due to the inertia of that liquid. |
| Amplitude | Maximum value of a sinusoidal quantity. |
| Angular Frequency | Frequency multiplied by 2π ; expressed in radians per unit time. |
| Antinode | An antinode is a point in a mode shape where the amplitude is a maximum. The opposite of a node. |

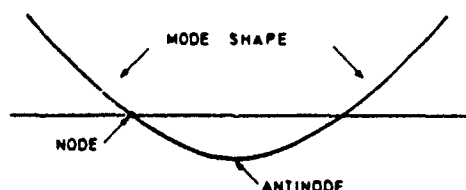


Figure A-1. Mode, Node, and Antinode

| | |
|------------------|---|
| Balancing | A procedure for adjusting the mass distribution of a rotor so that the forces on the bearings that occur once-per-revolution are reduced or controlled. |
| Band-Pass Filter | A band-pass filter is a wave filter that passes only the frequency components between a lower |

| | |
|---|---|
| | cutoff frequency greater than zero and a finite upper cutoff frequency. |
| Beats | Beats are periodic variations in amplitude that result from the superposition of two simple harmonic quantities of different frequencies f_1 and f_2 . They involve the periodic increase and decrease of amplitude at the beat frequency ($f_1 - f_2$). See Section 8.3.1. |
| Blade Frequency (Blade Rate) | The frequency at which propeller blades pass a fixed point. |
| Cavitation | The formation of local cavities in a liquid as a result of flash vaporization caused by the reduction of static pressure. As the pressure is increased the vapor liquifies again, and the resulting collapse of the cavity often causes damaging local pressures. |
| Circular Frequency | Frequency multiplied by 2π ; expressed in radians per unit time. |
| Complex Vibration | Vibration whose frequency components are sinusoids not harmonically related to one another. |
| Coulomb Damping (Dry Friction Damping) | Coulomb damping is the dissipation of energy that occurs when sliding surfaces in a vibrating system are resisted by a force whose magnitude is a constant independent of displacement and velocity. |
| Coupled Modes | Coupled modes are modes of vibration which influence one another because of energy transfer from one mode to the other. |
| Critical Damping | Critical damping is the minimum viscous damping that will allow a displaced system to return to its initial position without oscillation. |
| Critical Speed | Critical speed is a speed of a rotating system that corresponds to a resonance frequency of the system. |
| Cycle | A cycle is the complete sequence of values of a periodic quantity that occur during a period. |
| Damped Natural Frequency | The damped natural frequency is the frequency of free vibration of a damped linear system. |

| | |
|--|--|
| Damping | Damping is the dissipation of energy in a vibrating system. |
| Damping Ratio | The damping ratio for a system with viscous damping is the ratio of actual damping coefficient to the critical damping coefficient. |
| dBA Noise Level | <p>Sound level is a weighted sound pressure level, obtained by the use of metering characteristics and the weightings A, B, or C specified in American Standard Sound Level Meters for Measurement of Noise and Other Sounds, Z 24.3-1944. The weighting employed must always be stated. The reference pressure is 0.0002 microbar.</p> <p>When using the Sound Level Meter on the A weighting, the quantity obtained is referred to as dBA. This is the most commonly used scale on the SL meter since it has been found to agree fairly well with man's perception of sound.</p> |
| Degrees-of-Freedom | The number of degrees-of-freedom of a mechanical system is equal to the minimum number of independent coordinates required to define completely the positions of all parts of the system at any instant of time. (A rigid body has 6 degrees-of-freedom; three translations and three rotations.) |
| Displacement | Displacement is the change of position of a body and is usually measured from the mean position or position of rest. In general, it can be represented by a rotation, a translation, or both. |
| Double Amplitude | The peak-to-peak value or amplitude of an oscillating quantity is the algebraic difference between the extremes of the quantity. |
| Driving Point Impedance | The Impedance involving the ratio of force to velocity when both are measured at the same point and in the same direction. |
| Dynamic Vibration Absorber (Tuned Damper) | An auxiliary mass-spring system designed so that its vibrations are out of phase with those of the structure to which it is attached. The out-of-phase vibrations result in opposing forces on the structure and reduce its vibration. |

| | |
|--|--|
| Equivalent Viscous Damping | Equivalent viscous damping is a value of viscous damping assumed for the purpose of analysis of a vibratory motion, such that the dissipation of energy per cycle at resonance is the same for either the assumed or actual damping force. |
| Excitation (Stimulus) | Excitation is external forces or moments applied to a system that causes the system to move or vibrate. |
| Filter | A device for separating waves on the basis of their frequency. It introduces relatively small insertion loss to waves in one or more frequency bands and relatively large insertion loss to waves of other frequencies. |
| Flutter | A coupled torsional and lateral vibration of a ship's control surface which is aggravated by the lift force on the surface caused by the flow of water past it. |
| Flexural Hull Modes | Vertical and horizontal (athwartships) modes of hull vibration. |
| Forced Oscillation (Forced Vibration) | The oscillation of a system is forced if the response is imposed by the excitation. If the excitation is periodic and continuing, the oscillation is steady-state. |
| Fraction of Critical Damping | The fraction of critical damping for a system with viscous damping is the ratio of actual damping coefficient to the critical damping coefficient. |
| Free Vibration | Vibration that occurs in the absence of forced vibration. |
| Frequency | The frequency of a function periodic in time is the reciprocal of the period. The unit is the cycle per unit time. |
| Frequency Analyzer | A device that resolves an oscillating signal into its frequency components. The output is usually a plot of amplitude or energy versus frequency. |
| Fundamental Frequency | The fundamental frequency of an oscillating system is the lowest natural frequency. |
| Fundamental Mode of Vibration | The fundamental mode of vibration of a system is the mode having the lowest natural frequency. |

| | |
|-------------------------|--|
| g | The quantity g is the acceleration produced by the force of gravity. $g = 980.7 \text{ cm/sec}^2 = 386.1 \text{ in/sec}^2 = 32.17 \text{ ft/sec}^2$. |
| Harmonic | A harmonic is a sinusoidal quantity have a frequency that is an integral multiple of the frequency to which it is related. |
| Harmonic Motion | A harmonic motion is a motion such that the displacement is a sinusoidal function of time. |
| Harmonic Response | The periodic response of a vibrating system exhibiting the characteristics of resonance at a frequency that is a multiple of the excitation frequency. |
| Heave | The rigid body translation of a ship along its vertical axis. |
| High-Pass Filter | A high-pass filter is a wave filter that passes only the frequency components from some cutoff frequency, not zero, up to infinite frequency. |
| Hog | The distortion of a ship's hull due to greater buoyant forces near the middle than near the bow or stern. |
| Hull Pressure Amplitude | Magnitude of propeller-induced hull surface pressure. |
| Hysteresis Damping | Damping in which the dissipated energy is the difference between the energy used in deforming the parts of a mechanical system and the energy of the internal restoring forces. |
| Impedance | An impedance is the ratio of two quantities that represent a force-like and velocity-like quantity, respectively. (Examples of force-like quantities are: force, sound pressure, voltage, temperature, electric field strength. Examples of velocity-like quantities are: velocity, volume velocity, current, heat flow, magnetic flux). |
| Isolation | Isolation is a reduction in the capacity of a system to respond to an excitation attained by the use of a resilient support. |
| Lateral Vibration | Vibration in which a structure is bent along its length in the horizontal plane. |

| | |
|---|---|
| Line Spectrum | A line spectrum is a spectrum whose components occur at a number of discrete frequencies. |
| Linear Damping | Damping due to a damping force that is proportional to velocity. |
| Linear System | A system is linear if the response of every element in the system is proportional to the excitation. |
| Logarithmic Decrement | The logarithmic decrement is the natural logarithm of the ratio of the amplitudes of any two successive cycles in the decay of a single-frequency oscillation. |
| Longitudinal Vibration | Vibration in which a structure is alternately foreshortened and lengthened. |
| Low-Pass Filter | A low pass filter is a wave filter that passes only the frequency components from zero frequency up to some cutoff frequency, not infinite. |
| Lumping | The practice of considering the mass of complicated structure to be a series of point masses approximating the original mass distribution. |
| Mass | A measure of the inertia of a body, the ratio of force applied to the body to the resulting acceleration. At sea level mass is weight divided by acceleration due to gravity (32.2 ft./sec.^2 or 386.4 in./sec.^2). |
| Mechanical Impedance | Impedance when the quantities involved are force and velocity. |
| Mechanical Shock | A nonperiodic excitation (a motion of the foundation or an applied force) of a mechanical system that is characterized by suddenness and severity and usually causes significant relative displacements in the system. |
| Mechanical System | A mechanical system is an aggregate of matter comprising a defined configuration of mass, mechanical stiffness, and damping. |
| Mode Shape | Relative amplitudes of the parts of a system when vibrating at a natural frequency. |
| Mode of Vibration (Sometimes "Mode") | In a system undergoing vibration, a mode of vibration is a characteristic pattern assumed by the system, in which the motion of every particle |

| | |
|--|---|
| | is simple harmonic with the same frequency. Two or more modes may exist concurrently in a multiple degree-of-freedom system. |
| Modulus of Elasticity | Same as Young's Modulus. |
| Natural Frequency | Natural frequency is the frequency of free oscillation of a system. For a multiple degree-of-freedom system, the natural frequencies are the frequencies of the normal modes of vibration. |
| Node | A node is a point in a mode shape that has essentially zero amplitude. |
| Nominal Bandwidth | The nominal bandwidth of a filter is the difference between the nominal upper and lower cutoff frequencies. The difference may be expressed (1) in cycles per second; (2) as a percentage of the pass-band center frequency, or (3) as the interval between the upper and lower nominal cutoffs in octaves. |
| Nominal Upper and Lower Cutoff Frequencies | The nominal upper and lower cutoff frequencies of a filter pass-band are those frequencies above and below the frequency of maximum response of a filter at which the response to a sinusoidal signal is 3 dB below the maximum response. |
| Nonlinear Damping | Nonlinear damping is damping due to a damping force that is not proportional to velocity. |
| Normal Mode of Vibration | A normal mode of vibration is a mode of free vibration of an undamped system. In general, any composite motion of the system is a summation of its normal modes. |
| Oscillation | Oscillation is the variation, with time, of the magnitude of a quantity with respect to a specified reference when the magnitude is alternately greater and smaller than the reference. |
| Order of Vibration | In the vibration of rotating machinery, it is generally found that exciting torques and forces occur at the same frequency as the rotational speed, or at multiples of this frequency. The ratio of the frequency of vibration to the rotational speed is the order of vibration. |
| Peak Amplitude | Maximum value of a sinusoidal quantity (displacement, velocity, acceleration, pressure) |

| | |
|--|---|
| Peak-to-Peak Amplitude | Algebraic difference between extreme values of a sinusoidal quantity. |
| Peak-to-Peak Value (Double Amplitude) | The peak-to-peak value or amplitude of an oscillating quantity is the algebraic difference between the extremes of the quantity. |
| Period | The period of a periodic quantity is the smallest increment of time for which the function repeats itself. |
| Periodic Quantity | A periodic quantity is an oscillating quantity whose values recur for certain increments of time. |
| Phase of a Periodic Quantity | The phase of a periodic quantity, for a particular value of time, is the fractional part of a period through which time has advanced, measured from an arbitrary reference. (The arbitrary reference is generally so chosen that the fraction is less than unity. In case of a simple harmonic quantity the reference is often taken as the last previous passage through zero from the negative to positive direction.) |
| Phase Angle | Phase measured in degrees or radians; 360° or 2π radians correspond to one period. (Commonly used to indicate the difference in phase between two sinusoidal quantities having the same frequency.) |
| Pitch | Rigid body rotation of a ship about its athwartships axis. |
| Poisson's Ratio | The ratio of lateral unit strain to longitudinal unit strain, under the condition of uniform and uniaxial longitudinal stress within the proportional limit. |
| Pitch | The theoretical advance of a screw propeller along its axis in one revolution. |
| Q (Quality Factor) | <p>The quantity Q is a measure of the sharpness of resonance in a vibratory system having a single degree-of-freedom. In a mechanical system, this quantity is equal to one-half the reciprocal of the damping ratio. It is commonly used only with reference to a lightly damped system, and is then approximately equal to the following:</p> <ol style="list-style-type: none"> 1. Transmissibility at resonance. 2. π/δ where δ is the logarithmic decrement. |

3. $f_r/\Delta f$ where f_r is the resonant frequency and Δf is the bandwidth between the half-power points.

Random Vibration

Vibration whose instantaneous magnitude is not specified for any given instant of time. The instantaneous magnitudes of a random vibration are specified only by probability distribution functions giving the probable fraction of the total time that the magnitude lies within a specified range. Random vibration contains no periodic or quasi-periodic constituents.

Resonance

Resonance of a system in forced oscillation exists when any change, however small, in the frequency of excitation causes a large decrease in the response of the system. It occurs when the frequency of excitation coincides with a system natural frequency.

Resonant Frequency

A frequency at which resonance exists.

Response

The response of a system is the motion resulting from an excitation.

RMS Value (RMS Amplitude)

The RMS (root-mean-square) value of a function over an interval is the square root of the average of the square of all points in the function over that interval. In the case of a pure sinusoid the RMS value is the peak amplitude divided by $\sqrt{2}$.

Roll

Rigid body rotation of a ship about its longitudinal axis.

Sag

The distortion of ship's hull due to greater buoyant forces near the bow and stern than near the middle.

Self-Induced (Self-Excited) Vibration

The vibration of a mechanical system is self-induced if it results from conversion, within the system, of nonoscillatory excitation to oscillatory excitation. Examples of nonoscillatory excitation are unbalance and flow forces.

Shear Modulus of Elasticity (G)

A measure of shear strength of materials. For structural steel, $G = 7.72 \times 10^5$ ton/ft², 5.36×10^3 ton/in², or 12×10^6 lb/in².

| | |
|------------------------------|---|
| Shock Absorber | A shock absorber is a device for the dissipation of energy to reduce the response of a mechanical system to applied shock. |
| Shock Isolator (Shock Mount) | A shock isolator is a resilient support that tends to isolate a system from applied shock. |
| Simple Harmonic Motion | A simple harmonic motion is a motion such that the displacement is a sinusoidal function of time. |
| Single Amplitude | The measure of vibratory motion from a mean (zero) position to an extreme. |
| Sinusoidal Motion | A sinusoidal motion is a motion such that the displacement is a sinusoidal function of time. |
| Slamming | Slamming occurs when the ship bottom near the bow rises out of the sea and then hits the water, resulting in transient vibrations of the hull. |
| Snubber | A snubber is a device used to increase the stiffness of an elastic system (usually by a large factor) whenever the displacement becomes larger than a specified amount. |
| Spectrum | The spectrum of a function of time is a description of its resolution into components, each of different frequency and (usually) different amplitude and phase. |

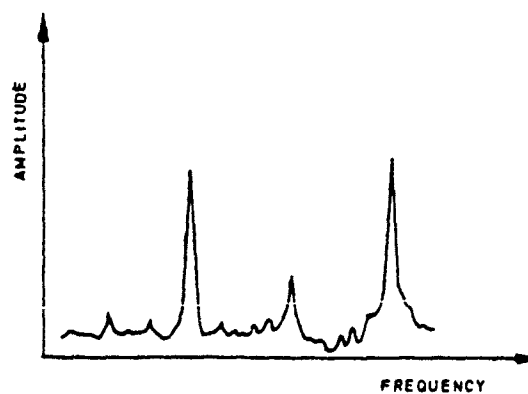


Figure A-2. Frequency Spectrum

Spectrum Density

The limiting mean-square value of a quantity per unit bandwidth, i.e., the limit of the mean-square value in a given rectangular bandwidth divided by the bandwidth, as the bandwidth approaches zero.

Steady-State Vibration

Steady-state vibration exists in a system if the motion of each particle is a continuing periodic quantity.

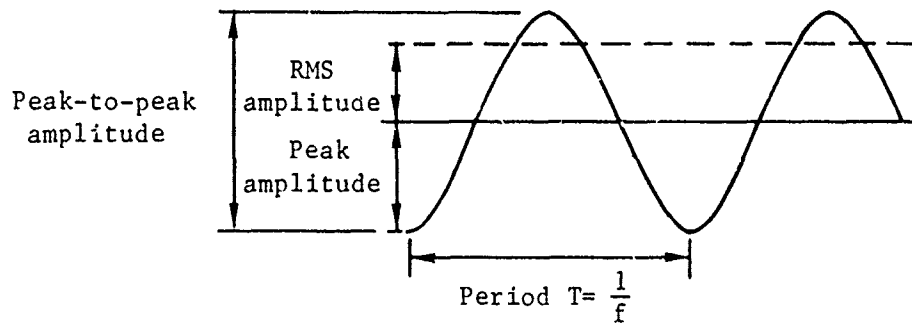


Figure A-3. Steady-State Vibration Amplitude

Stiffness (Spring Constant)

Stiffness is the ratio of change of force (or torque) to the corresponding change in translational (or rotational) displacement of an elastic element.

Subcritical

Running below the critical speed.

Subharmonic

A subharmonic is a sinusoidal quantity having a frequency that is an integral submultiple of the fundamental frequency of a periodic quantity to which it is related.

Supercritical

Running above the critical speed.

Subharmonic Response

Subharmonic response is the periodic response of a mechanical system exhibiting the characteristic of resonance at a frequency that is a submultiple of the frequency of the periodic excitation.

Surge

Rigid body translation of a ship along its longitudinal axis.

Sway

Rigid body translation of a ship along its athwartship axis.

| | |
|----------------------------|--|
| Torsional Vibration | Vibration in which a structure is twisted. |
| Transducer (Pickup) | A device which converts shock or vibratory motion into an optical, a mechanical, or most commonly into an electrical signal that is proportional to a parameter (such as acceleration) of the experienced motion. |
| Transient Vibration | Transient vibration is temporarily sustained vibration of a mechanical system. It may consist of forced or free vibration or both. |
| Transmissibility | Transmissibility is the ratio of the response amplitude of a system in steady-state forced vibration to the excitation amplitude expressed in the same units. The ratio may be one of forces, displacements, velocities, or accelerations. |
| Transmission Loss | Transmission loss is the reduction in the magnitude of some characteristic of a signal, between two stated points in a transmission system. |
| Tuned Damper | An auxiliary mass-spring system designed so that its vibrations are out of phase with those of the structure to which it is attached. The out-of-phase vibrations result in opposing forces on the structure and reduce its vibration. |
| Uncoupled Mode | An uncoupled mode of vibration is a mode that can exist in a system concurrently with and independently of other modes. |
| Undamped Natural Frequency | The undamped natural frequency of a mechanical system is the frequency of free vibration resulting from only elastic and inertial forces of the system. |
| Vdb Velocity Level | The velocity level, in decibels of a sound, is 20 times the logarithm to the base 10 of the ratio of the particle velocity of the sound to the reference particle velocity. $V_{db} = 20 \log \frac{V_1}{V_2}, \text{ where } V_2 = 10^{-6} \text{ cm/sec}$ |
| Velocity | Velocity is the time rate of change of displacement. |
| Vibration | Vibration is an oscillation of a mechanical system. |

| | |
|---------------------|---|
| Vibration Isolator | A vibration isolator is a resilient support that tends to isolate a system from steady-state excitation. |
| Vibratory System | Any (structural or mechanical) system containing mass and elasticity. |
| Vibrometer | A vibration amplitude indicating device. |
| Virtual Mass | The apparent increase in mass of a body vibrating in water due to the inertia of water. |
| Viscous Damping | Viscous damping is the dissipation of energy that occurs when a vibrating system is resisted by forces whose magnitudes are proportional to the velocities of the system. |
| Wake | Description of the velocity of water flow behind a ship. |
| Whipping | Lateral vibration of a shaft system, or of a hull girder due to its encounter with waves. |
| Whirling Shaft | Vibration of a shaft in which the shaft center describes a circular or elliptical path in a plane normal to the axis of the shaft. |
| Yaw | Rigid body rotation of a ship about its vertical axis. |
| Young's Modulus (E) | The rate of change of unit tensile or compressive stress with respect to unit tensile or compressive strain for the condition of uniaxial stress within the proportional limit. |

$$E = \frac{(\text{load}) \times (\text{length})}{(\text{cross-sectional area}) \times (\text{extension})}$$

For structural steel $E = 1.93 \times 10^6 \frac{\text{ton}}{\text{ft}^2}$,

$13.4 \times 10^3 \text{ ton/in}^2$, or $3 \times 10^7 \text{ lb/in}^2$.

A.1.2 Notation Used in Section A.0

The notation given in Table A-1 is used in discussing mechanical vibrating systems in Appendix A. In other sections the notation is usually taken from the original source and is defined in that section.

Table A-1. Notation for Appendix A

| <u>Symbol</u> | <u>Definition</u> | <u>Units</u> | <u>Dimensions in Force-Length-Time System</u> |
|----------------|--|---------------------|---|
| A | Cross-sectional area | in ² | ℓ ² |
| A _i | Coefficient for i th mode | None | None |
| a | Linear acceleration | in/sec ² | ℓt ⁻² |
| a _o | Amplitude of sinusoidal linear acceleration | in/sec ² | ℓt ⁻² |
| C _i | Constants | None | None |
| c | Linear damping constant | lb-sec/in | fℓ ⁻¹ t |
| c _c | Critical damping coefficient | lb-sec/in | fℓ ⁻¹ t |
| c _r | Rotational damping constant | lb-in-sec/rad | fℓt |
| D | Diameter of solid shaft | in | ℓ |
| d | Diameter of wire in spring | in | ℓ |
| D _i | Inside diameter of hollow shaft | in | ℓ |
| D _o | Outside diameter of hollow shaft | in | ℓ |
| D _s | Outside diameter of spring | in | ℓ |
| E | Modulus of elasticity (about 30 x 10 ⁶ for steels) | lb/in ² | fℓ ⁻² |
| e | 2.71828 | None | None |
| F | Magnification Factor, x _o /x _{st} | None | None |
| F _c | Friction force | lbs | f |
| f | Frequency | Hz | t ⁻¹ |
| f _i | i th natural frequency | Hz | t ⁻¹ |
| f _n | Undamped natural frequency | Hz | t ⁻¹ |
| f _r | Resonant frequency | Hz | t ⁻¹ |

Table A-1. Notation for Appendix A (Continued)

| <u>Symbol</u> | <u>Definition</u> | <u>Units</u> | <u>Dimensions in Force-Length-Time System</u> |
|---------------|--|-------------------------|---|
| G | Shear modulus of elasticity (about 11.5×10^6 for steels) | lb/in ² | $f\ell^{-2}$ |
| g | Acceleration due to gravity | in/sec ² | ℓt^{-2} |
| H | Height or thickness | in | |
| I | Area moment of Inertia of cross-section | in ⁴ | ℓ^4 |
| I_p | Polar area moment of Inertia of cross-section | in ⁴ | ℓ^4 |
| J | Polar mass moment of Inertia | lb-in-sec ² | $f\ell t^2$ |
| J_e | Polar mass moment of Inertia of engine | lb-in-sec ² | $f\ell t^2$ |
| J_p | Polar mass moment of Inertia of propeller | lb-in-sec ² | $f\ell t^2$ |
| k | Linear spring constant | lb/in | $f\ell^{-1}$ |
| k_t | Torsional spring constant | lb-in/rad | $f\ell$ |
| L | Length of beam, string, or shaft | in | ℓ |
| L_f | Length as shown | in | ℓ |
| L_o | Distance from propeller to center of propeller bearing | in | ℓ |
| L_s | Distance from center of propeller bearing to line bearing | in | ℓ |
| M | Torque | lb-in | $f\ell$ |
| m | Mass | lb-sec ² /in | $f\ell^{-1}t^2$ |
| m_b | Mass of beam | lb-sec ² /in | $f\ell^{-1}t^2$ |
| m_{es} | Effective mass of shaft (.38 x mass) | lb-sec ² /in | $f\ell^{-1}t^2$ |

Table A-1. Notation for Appendix A (Continued)

| <u>Symbol</u> | <u>Definition</u> | <u>Units</u> | <u>Dimensions in Force-Length-Time System</u> |
|---------------|--|-------------------------|---|
| m_p | Propeller mass + 10% for entrained water (lateral vibration) | lb-sec ² /in | $f\ell^{-1}t^2$ |
| N | Number of degrees-of-freedom | None | None |
| n | Number of coils in a spring | None | None |
| P | Force | lbs | f |
| P_o | Amplitude of sinusoidal force | lbs | f |
| P_s | Tension in string | lbs | f |
| P_t | Force transmitted to the foundation | lbs | f |
| Q | Quality factor (F at resonance) | None | None |
| q | Number of cycles | None | None |
| T | Period | sec | t |
| t | Time | sec | t |
| Tr | Transmissibility (force or motion) | None | None |
| u | Linear displacement of foundation | in | ℓ |
| u_o | Amplitude of sinusoidal linear displacement of foundation | in | ℓ |
| v | Linear velocity | in/sec | ℓt^{-1} |
| v_o | Amplitude of sinusoidal linear velocity | in/sec | ℓt^{-1} |
| W | Width | in | ℓ |
| w | Weight | lb | f |
| x | Linear displacement | in | ℓ |

Table A-1. Notation for Appendix A (Concluded)

| <u>Symbol</u> | <u>Definition</u> | <u>Units</u> | <u>Dimensions in Force-Length-Time System</u> |
|---------------|---|--------------------------------------|---|
| x_i | Amplitude of the i th cycle, or of i th mass | in | ℓ |
| x_m | Amplitude of major component in beat | in | ℓ |
| x_n | Amplitude of minor component in beat | in | ℓ |
| x_o | Amplitude of sinusoidal linear displacement | in | ℓ |
| x_{st} | Linear static deflection | in | ℓ |
| \dot{x} | Linear velocity | in/sec | ℓt^{-1} |
| \ddot{x} | Linear acceleration | in/sec ² | ℓt^{-2} |
| δ | Logarithmic decrement | None | None |
| θ | Phase angle | rad | None |
| μ | Mass per unit length of beam or string | lb-sec ² /in ² | $f \ell^{-2} t^2$ |
| ν | Poisson's ratio | None | None |
| ϕ | Angular displacement | rad | None |
| ρ | Weight density | lb/in ³ | $f \ell^{-3}$ |
| ω | Angular frequency | rad/sec | t^{-1} |
| ω_n | Angular natural frequency (undamped) | rad/sec | t^{-1} |
| ω_d | Damped angular natural frequency | rad/sec | t^{-1} |

A.1.3 Metric Conversion Factors

Most of the units in the body of this report are in the English System. Table A-2 is provided for converting from English in metric units. If the metric units are known, dividing by the factors given will give the English units.

Table A-2. Metric Conversion Factors

| <u>Symbol</u> | <u>When You Know</u> | <u>Multiply By</u> | <u>To Find</u> | <u>Symbol</u> |
|-----------------------------|--------------------------|--------------------|------------------------|-----------------|
| <u>Length, Displacement</u> | | | | |
| in | Inches | 25.4 | Millimeters | mm |
| mils | 1/1000 inches | 0.0254 | Millimeters | mm |
| ft | Feet | 0.3048 | Meters | m |
| Fa | Fathom | 1.8288 | Meters | m |
| <u>Area</u> | | | | |
| in ² | Square inches | 645.16 | Square millimeters | mm ² |
| ft ² | Square feet | 0.09290304 | Square meters | m ² |
| <u>Volume</u> | | | | |
| in ³ | Cubic inches | 1638.706 | Cubic millimeters | mm ³ |
| ft ³ | Cubic feet | 0.02831685 | Cubic meters | m ³ |
| gal | Gallons | 0.003785412 | Cubic meters | m ³ |
| gal | Gallons | 3.785412 | Liters | |
| <u>Linear Velocity</u> | | | | |
| in/s | Inches per second | 25.4 | Millimeters per second | mm/s |
| mils/s | 1/1000 inches per second | 0.0254 | Millimeters per second | mm/s |
| ft/s | Feet per second | 0.3048 | Meters per second | m/s |
| kn | Knots | 0.514444 | Meters per second | m/s |

Table A-2. Metric Conversion Factors (Continued)

| <u>Symbol</u> | <u>When You Know</u> | <u>Multiply By</u> | <u>To Find</u> | <u>Symbol</u> |
|--------------------------------------|-----------------------------|--------------------|--------------------------------|-------------------|
| <u>Frequency (Circular Velocity)</u> | | | | |
| Hz | Cycles per second | 6.283185 | Radians per second | rd/s |
| Hz | Cycles per second | 60.0 | Cycles per minute | CPM |
| <u>Acceleration</u> | | | | |
| in/s ² | Inches per second squared | .00259 | Acceleration due to gravity | g |
| ft/s ² | Feet per second squared | .03108 | Acceleration due to gravity | g |
| g | Acceleration due to gravity | 9807 | Millimeters per second squared | mm/s ² |
| g | Acceleration due to gravity | 9.807 | Meters per second squared | m/s ² |
| in/s ² | Inches per second squared | 25.4 | Millimeters per second squared | mm/s ² |
| ft/s ² | Feet per second squared | 0.3048 | Meters per second squared | m/s ² |
| <u>Force</u> | | | | |
| lb | Pounds | 4.448222 | Newton | N |
| LT | Long ton | 9964.017 | Newton | N |
| LT | Long ton | 1.016047 | Tonne | T |
| <u>Energy, Work</u> | | | | |
| BTU | British Thermal Unit | 1055.056 | Joules | J |
| ft-lb | Foot pounds | 1.355818 | Joules | J |

Table A-2. Metric Conversion Factors (Concluded)

| <u>Symbol</u> | <u>When You Know</u> | <u>Multiply By</u> | <u>To Find</u> | <u>Symbol</u> |
|-------------------------|---------------------------|--------------------|--|------------------------|
| <u>Mass</u> | | | | |
| lb | Pound | 0.4535924 | Kilogram | kg |
| LT | Long ton | 1016.047 | Kilogram | kg |
| <u>Power</u> | | | | |
| HP | Horsepower | 745.6999 | Watts | W |
| HP | Horsepower | 0.9863 | Metric horsepower | hp |
| <u>Pressure, Stress</u> | | | | |
| psi | Pounds per square inch | 6,894.757 | Newton per square meter (or Pascal) | N/m ² Pa |
| ksi | Kips per square inch | 6,894.757 | Newton per square meter | N/m ² |
| LT/in ² | Long tons per square inch | 15,444.260 | Newton per square meter | N/m ² |
| <u>Angle</u> | | | | |
| ° | Degree | /180 | Radians | rad |
| <u>Temperature</u> | | | | |
| °F | Fahrenheit | (°F-32)/1.8 | Celsius | °C |

A.2 VIBRATION FUNDAMENTALS

This section utilizes a single degree-of-freedom system to present the basic theory of vibration. Parameters used in vibration analysis are discussed in relation to this system. An indication of how to approach a multiple degree-of-freedom system is included, but details are omitted. Equations are included for completeness, but an effort has been made to make this section understandable on the basis of text and curves only.

A.2.1 Harmonic Motion

A vibration in its general sense is a periodic motion. A plot of displacement against time may be simple harmonic motion as described in Figure A-4, or it may be of a complex nature. Any periodic motion can be expressed as a sum of harmonic motions.

If: T is period in seconds,

f is frequency in cycles per second, and

ω is angular frequency in radians per sec;

then the following is true of harmonic motion:

$$f = \frac{1}{T}$$

$$\omega = 2\pi f$$

$$T = \frac{2\pi}{\omega}$$

$$f = \frac{\omega}{2\pi}$$

Displacement is the easiest quantity to visualize and a sinusoidal displacement, x , can be mathematically represented by:

$$x = x_0 \sin \omega t$$

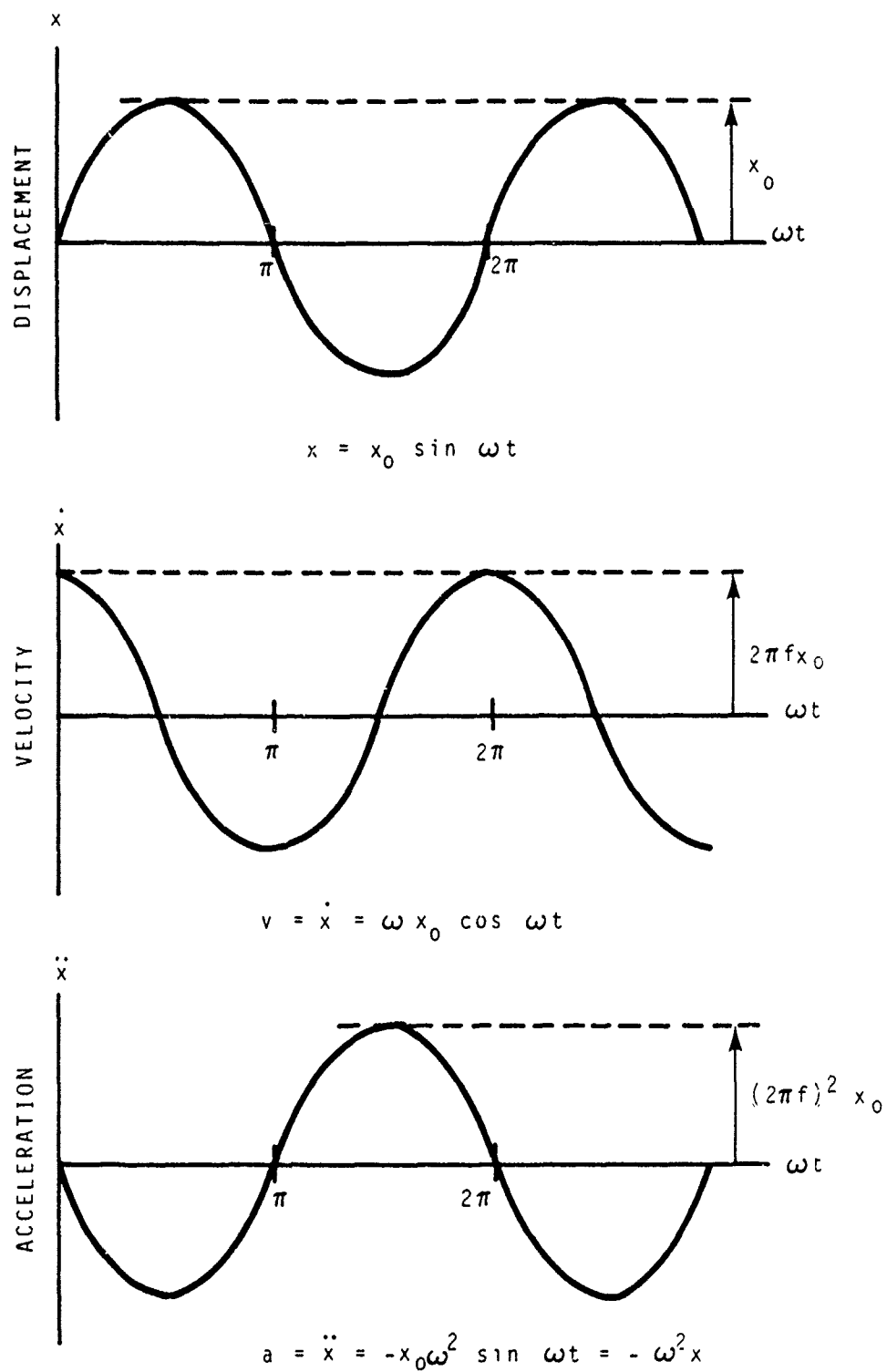


Figure A-4. Relationship Between Displacement, Velocity, and Acceleration in Simple Harmonic Motion

where x_0 is the displacement amplitude. The velocity, v , can be obtained by differentiating:

$$v = \frac{dx}{dt} = x_0 \omega \cos \omega t$$

The acceleration can be found by differentiating a second time:

$$a = \frac{dv}{dt} = -x_0 \omega^2 \sin \omega t$$

If v and a are the single amplitudes of velocity and acceleration, respectively, then:

$$v_0 = \omega x_0$$
$$a_0 = \omega^2 x_0$$

This shows that the displacement, velocity, and acceleration amplitudes of sinusoids differ only by factors of the frequency. This makes it possible to plot vibration amplitudes on a graph such as Figure A-5 which shows all three parameters.

A.2.2 Single Degree-of-Freedom System

A simplified translational single degree-of-freedom system is shown in Figure A-6. The parts of the system are:

1. A massless spring, labeled k , whose change in length is proportional to the force acting on it.
2. A viscous damper, labeled c , which dissipates energy. The applied force is proportional to the relative velocity of its connection points. (This does not accurately represent the damping that occurs in most physical systems.)
3. A mass, labeled m , whose acceleration is proportional to the applied force. It is constrained so that it moves vertically only.

A simplified rotational single degree-of-freedom system (Figure A-7) is analogous to the translational system. Table A-3 indicates the analogous quantities, and also the notation used in both systems. Since rotational vibration is analogous to translational vibration, only the latter will be considered in subsequent discussions.

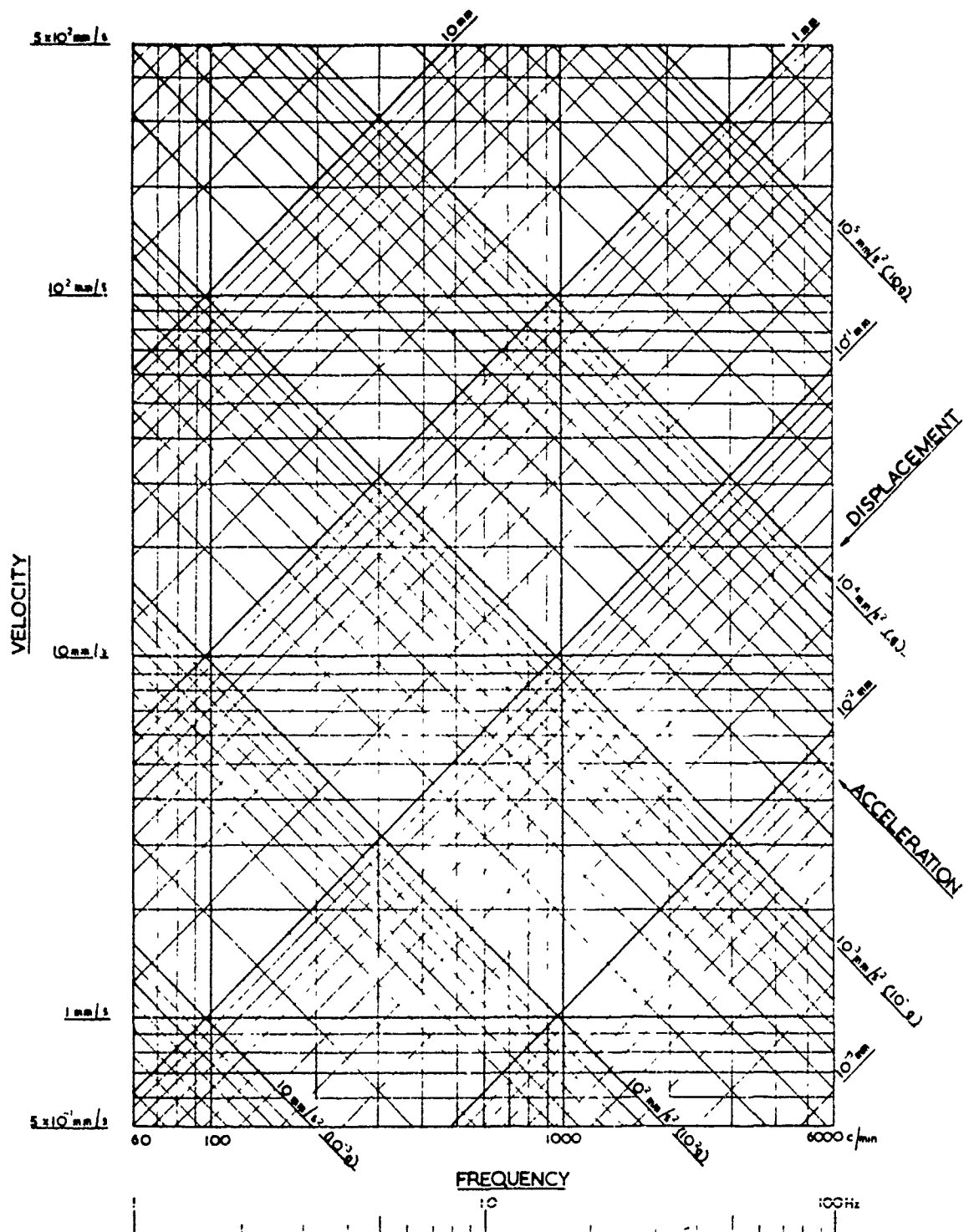


Figure A-5. Vibration Analysis Form Relating Displacement, Velocity, Acceleration, and Frequency

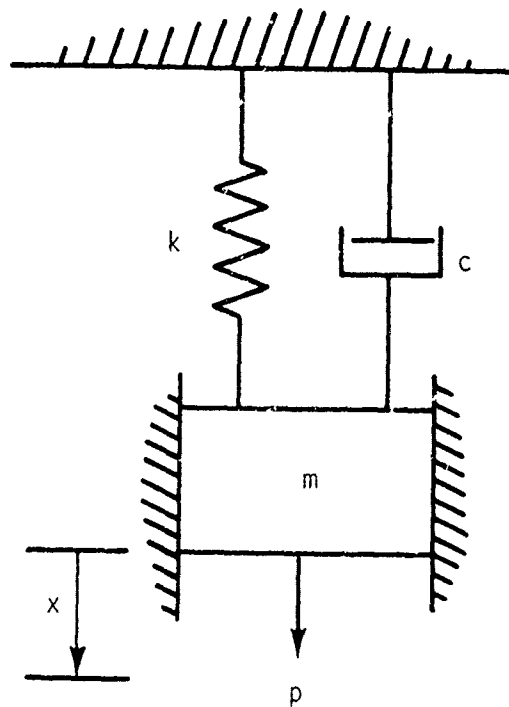


Figure A-6. Translational Single Degree-of-Freedom System

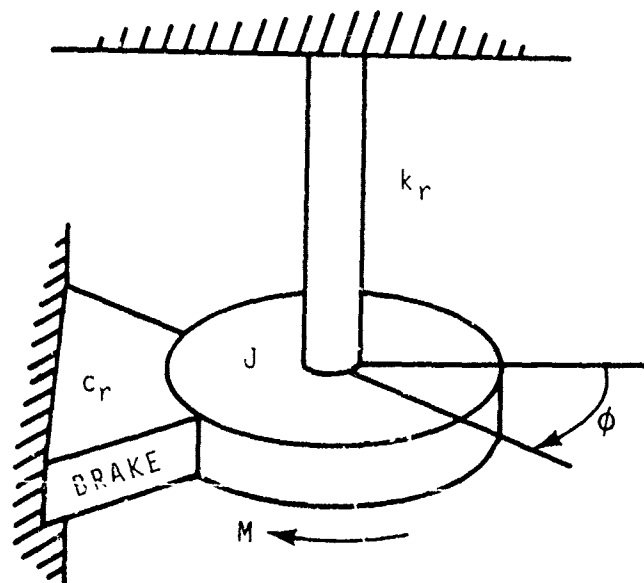


Figure A-7. Rotational Single Degree-of-Freedom System

Table A-3. Analogous Quantities in Translational and Rotational Systems

| Translational | | Rotational | |
|---|-----|---|-------|
| Linear Displacement | x | Angular Displacement | |
| Force Applied to Mass | P | Torque Applied to Mass | M |
| Spring Constant (restoring force per unit displacement) | k | Spring Constant (restoring torque per unit of angular displacement) | k_r |
| Damping Constant (damping force per unit velocity) | c | Damping Constant (damping torque per unit of angular velocity) | c_r |
| Mass (weight/g) | m | Mass Polar Moment of Inertia | J |

A.2.2.1 General Equation of Motion

If the force P acting on the mass is sinusoidal, $P = P_0 \sin \omega t$, equating all forces acting on the mass results in the equation of motion:

$$m\ddot{x} + c\dot{x} + kx = P_0 \sin \omega t \quad (\text{where } x = 0 \text{ is equilibrium position.})$$

A.2.2.2 Free Vibrations Without Damping

If vibration is caused by an initial condition such as displacing the mass from its equilibrium position, then allowing the system to oscillate free from external forces and damping effects, the equation reduces to

$$m\ddot{x} + kx = 0$$

The solution for this equation is a sinusoid with a constant amplitude:

$$x = C_1 \sin \sqrt{\frac{k}{m}} t + C_2 \cos \sqrt{\frac{k}{m}} t$$

where $\sqrt{\frac{k}{m}}$ is the angular natural frequency, ω_n . The natural frequency, f_n of the system is

$$f_n = \frac{\omega}{2\pi} = \frac{1}{2\pi} \sqrt{\frac{k}{m}}$$

C_1 and C_2 are determined by the initial conditions. The solution can also be written:

$$x = C_1 \sin \omega_n t + C_2 \cos \omega_n t = C_3 \sin (\omega_n t + \theta)$$

where $C_3 = \sqrt{C_1^2 + C_2^2}$ and $\theta = \tan^{-1}(\frac{C_2}{C_1})$. The angle θ is the phase angle.

A.2.3.3 Free Vibration with Viscous Damping

If, after an initial disturbance of the system shown in Figure A-6, the external force ceases to act, the equation becomes:

$$m\ddot{x} + c\dot{x} + kx = 0$$

The form of the solution of this equation depends on whether c is equal to, greater than, or less than the critical damping coefficient, c_c , where

$$c_c = 2\sqrt{km} = 2m\omega_n$$

The ratio c/c_c is defined as the fraction of critical damping.

If $c/c_c < 1$, the system is said to be underdamped, and the solution is:

$$x = e^{-ct/2m} (C_1 \sin \omega_n t + C_2 \cos \omega_n t) = C_3 e^{-ct/2m} \sin(\omega_n t + \theta)$$

where $C_3 = \sqrt{C_1^2 + C_2^2}$ and the phase angle, $\theta = \tan^{-1}(\frac{C_2}{C_1})$.

After an initial disturbance, an underdamped system continues to oscillate at a continuously reducing amplitude and is referred to as a decaying oscillation, as shown in Figure A-8, with the damped natural frequency which is slightly lower than the undamped:

$$\omega_d = \sqrt{\frac{k}{m} - \frac{c^2}{4m^2}} = \omega_n [1 - (\frac{c}{c_c})^2]^{1/2}$$

If $c/c_c > 1$ the system is said to be overdamped, and the mass will not oscillate but will tend to return to its equilibrium position gradually.

The amplitude of an underdamped system decreases with each cycle. The logarithmic decrement (δ) is the natural logarithm of the ratio of two successive amplitudes.

$$\delta = \log_e \frac{x_1}{x_{i+1}}$$

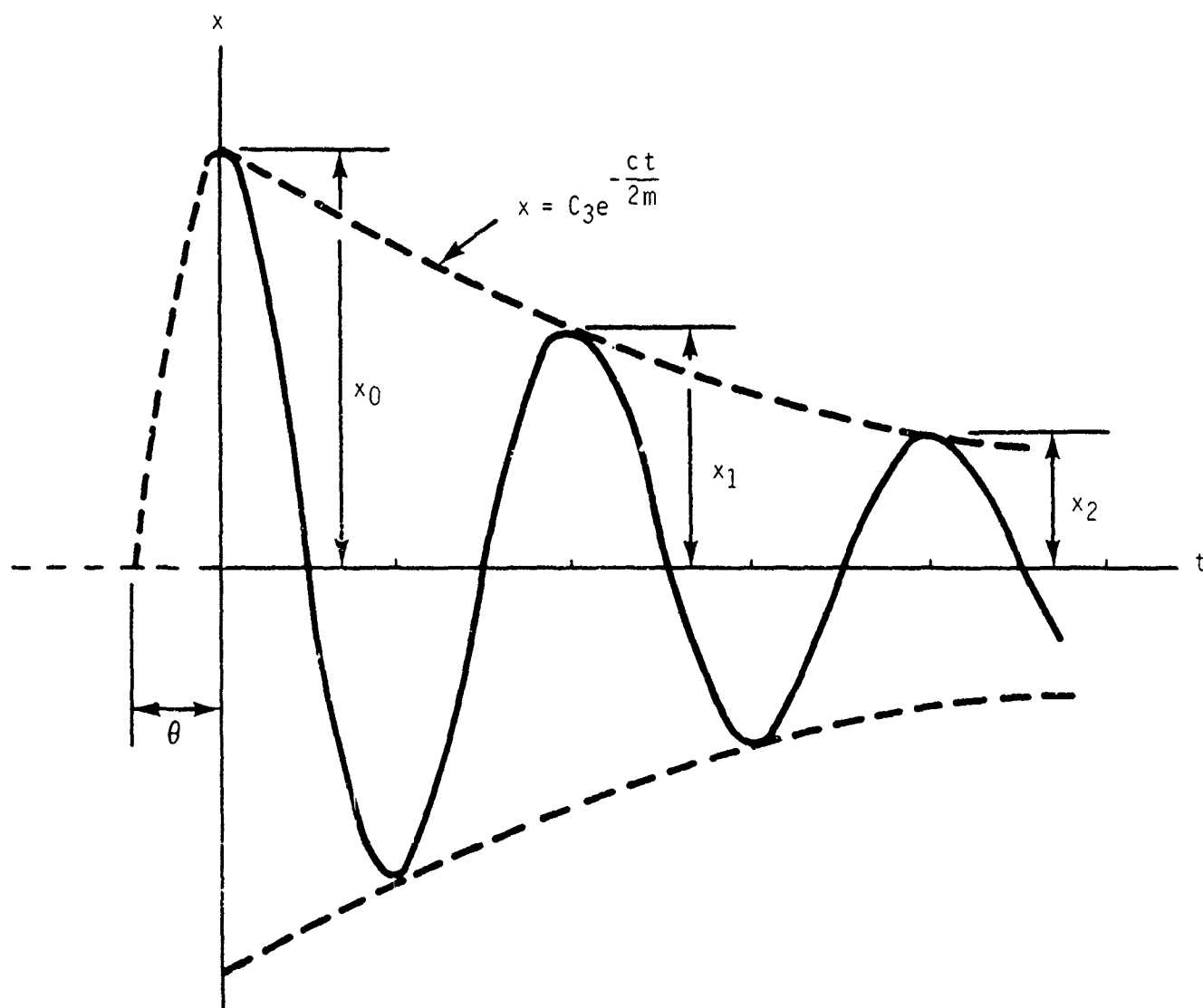


Figure A-8. Displacement as a Function of Time in a Free Underdamped Single Degree-of-Freedom System

or, for an interval of q cycles

$$\delta = \frac{1}{q} \log_e \frac{x_1}{x_{1+q}}$$

Other relationships for δ are:

$$\delta = \frac{\pi c}{m \omega_n} = \frac{2\pi \frac{c}{c_c}}{\sqrt{1 - \left(\frac{c}{c_c}\right)^2}}$$

For small damping

$$\delta \approx 2\pi \frac{c}{c_c}$$

A.2.2.4 Forced Vibration Without Damping

Forced vibration may result from the application of force to the mass or from the motion of the foundation.

When the sinusoidal force $P = P_o \sin \omega t$ is applied to the mass of the undamped single degree-of-freedom system, the equation of motion is

$$m\ddot{x} + kx = P_o \sin \omega t$$

The solution of this equation is

$$x = C_1 \sin \omega_n t + C_2 \cos \omega_n t + \frac{\left(\frac{P_o}{k}\right) \sin \omega t}{1 - \left(\frac{\omega}{\omega_n}\right)^2}$$

where $\omega_n = \sqrt{\frac{k}{m}}$. The first two terms ($C_1 \sin \omega_n t + C_2 \cos \omega_n t$) represent an oscillation at the undamped natural frequency ω_n . This oscillation gradually decays to zero in physical systems because of damping. The steady-state oscillation at the forcing frequency ω is given by the third term.

$$x = \frac{\left(\frac{P_o}{k}\right) \sin \omega t}{1 - \left(\frac{\omega}{\omega_n}\right)^2}$$

This can be written:

$$\frac{x_o}{x_{st}} = \frac{1}{1 - \left(\frac{\omega}{\omega_n}\right)^2}$$

where: x_o is the amplitude of x

x_{st} is $\frac{P_o}{k}$, the static deflection produced by a force P_o .

The ratio x_o/x_{st} is the magnification factor and is plotted as a function of ω/ω_n in Figure A-9. When $\omega/\omega_n < 1$, x_o/x_{st} is positive and when $\omega/\omega_n > 1$, x_o/x_{st} is negative. This is the result of force and motion being in phase for $\omega/\omega_n < 1$, but out of phase when $\omega/\omega_n > 1$. Normally the curve does not indicate phase and the negative part of the curve is drawn as shown by the dotted line. When $\omega = \omega_n$ a condition of resonance exists, and the amplitude increases continuously with time.

The force transmitted to the foundation is directly proportional to the spring deflection: $P_t = kx$. Substituting our previous expression for x and defining transmissibility, $Tr = P_t/P$, (force transmissibility)

$$Tr = \frac{1}{1 - \left(\frac{\omega}{\omega_n}\right)^2}$$

which is the same as the expression for x_o/x_{st} .

The equation of motion for the system of Figure A-6, with no force acting on the mass, but excited by a motion of the foundation, $u = u_o \sin \omega t$, is:

$$m\ddot{x} = -k(x - u_o \sin \omega t)$$

The solution is

$$x = C_1 \sin \omega_n t + C_2 \cos \omega_n t + \frac{u_o}{1 - \left(\frac{\omega}{\omega_n}\right)^2} \sin \omega t$$

where $\omega_n = \sqrt{\frac{k}{m}}$. The terms representing oscillation at the natural frequency decay, resulting in a steady-state oscillation:

$$x = \frac{u_o}{1 - \left(\frac{\omega}{\omega_n}\right)^2} \sin \omega t$$

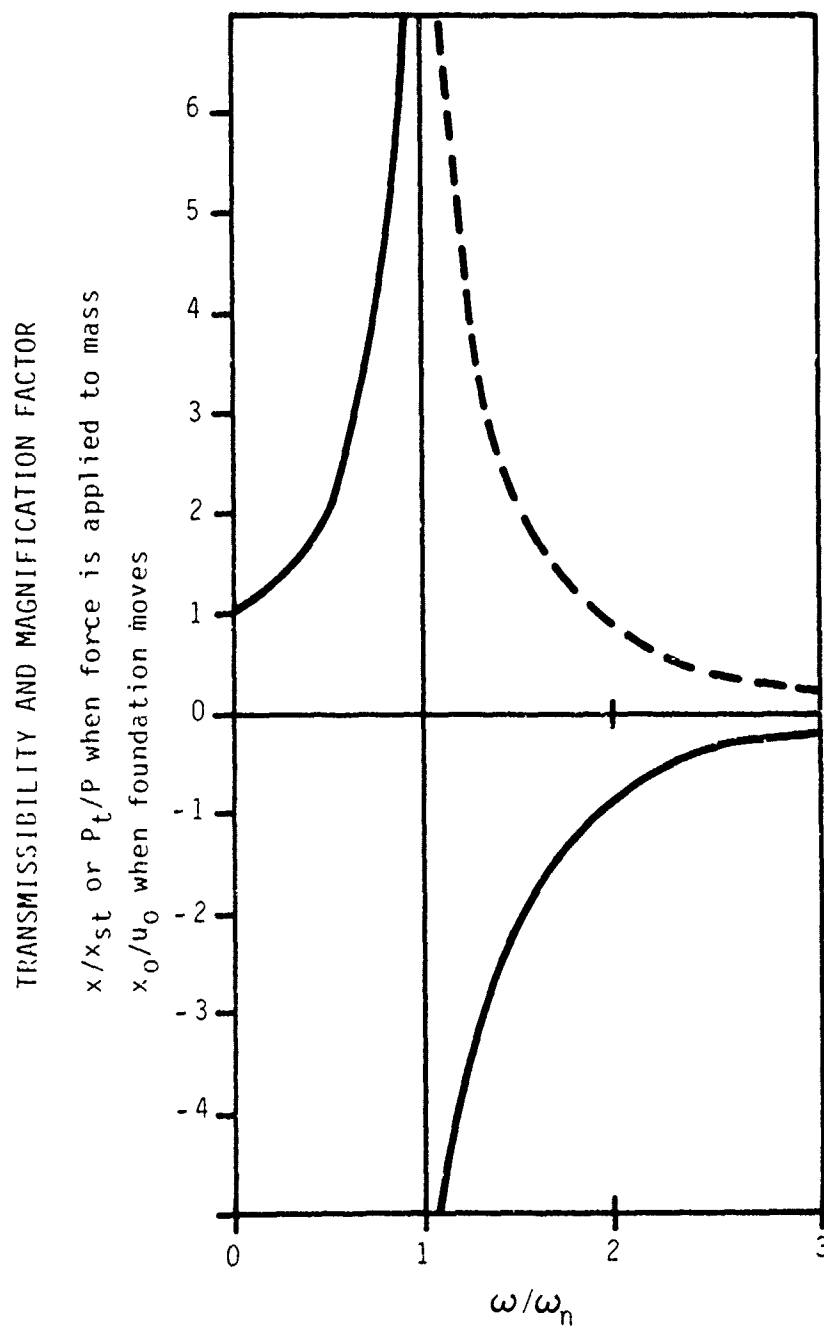


Figure A-9. Transmissibility and Magnification Factor as a Function of Frequency Ratio for a Forced, Undamped, Single Degree-of-Freedom System

The amplitude of this motion is

$$x_o = \frac{u_o}{1 - \left(\frac{\omega}{\omega_n}\right)^2}$$

Defining transmissibility as the ratio of amplitudes of the mass and foundation, $Tr = \frac{x_o}{u_o}$, (motion transmissibility).

$$Tr = \frac{1}{1 - \left(\frac{\omega}{\omega_n}\right)^2}$$

This shows that the magnification factor is the same as force transmissibility when a force is applied to the mass, and the same as motion transmissibility when foundation motion causes the vibration. For the undamped system all are given by Figure A-9.

A.2.2.5 Forced Vibration with Viscous Damping

Forced vibration may result from a force applied to the mass or from motion of the foundation. If the driving force $P = P_o \sin \omega t$ is applied to the mass, the equation of motion becomes

$$m\ddot{x} + c\dot{x} + kx = P_o \sin \omega t$$

The solution of this equation is

$$x = e^{-\frac{ct}{2m}} (C_1 \sin \omega_d t + C_2 \cos \omega_d t) + \frac{P_o \sin(\omega t + \theta)}{\sqrt{(c\omega)^2 + (k - m\omega^2)^2}}$$

where $\omega_d = \sqrt{\frac{k}{m} - \frac{c^2}{4m^2}}$ and $\theta = \tan^{-1}\left(\frac{c\omega}{k - m\omega^2}\right)$

The first term decays due to damping and the resulting steady-state amplitude is

$$x_o = \frac{P_o}{\sqrt{(c\omega)^2 + (k - m\omega^2)^2}} \quad \text{or} \quad \frac{\frac{P_o}{k}}{\sqrt{\left[1 - \left(\frac{\omega}{\omega_n}\right)^2\right]^2 + \left[2 \frac{c\omega}{c\omega_n}\right]^2}}$$

where P_0/k is static deflection, x_{st} produced by a force of P_0 .

The magnification factor, F , is the ratio x_0/x_{st} . Figures A-10 and A-11 plot the magnification factor and the phase angle against the frequency ratio for various values of c/c_c .

Force transmissibility is the ratio:

$$\frac{\text{force applied to the mass}}{\text{force in spring}}$$

This is slightly different than the magnification factor which applies to amplitudes. Transmissibility is shown in Figure A-12. If the system is excited by a motion of the base we may be interested in motion transmissibility which is the ratio:

$$\frac{\text{mass motion}}{\text{base motion}}$$

Motion transmissibility in this case is the same as force transmissibility for the forced case, and Figure A-12 still applies. Note that the transmissibility is less than one when the exciting frequency is greater than $\sqrt{2}$ times the natural frequency.

The amount of damping in a system is indicated by the quality factor, Q , which is the magnification factor at the undamped natural frequency.

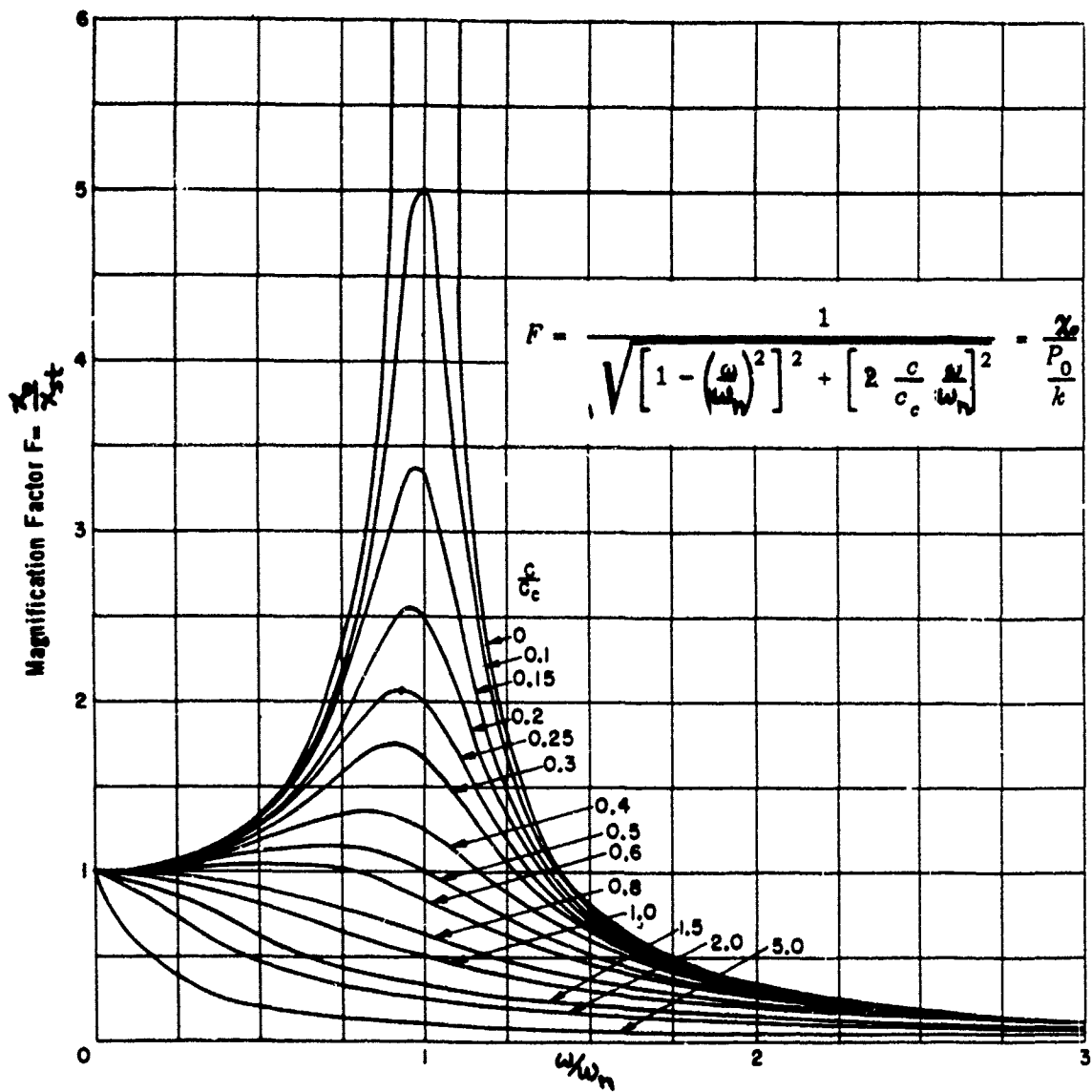


Figure A-10. Magnification Factor for a System of One Degree-of-Freedom with Viscous Damping Acted on by a Force $P_0 \sin \omega t$

$$F_{\max} \text{ occurs at } \frac{\omega}{\omega_n} = \sqrt{1 - 2 \frac{c}{c_c}^2}.$$

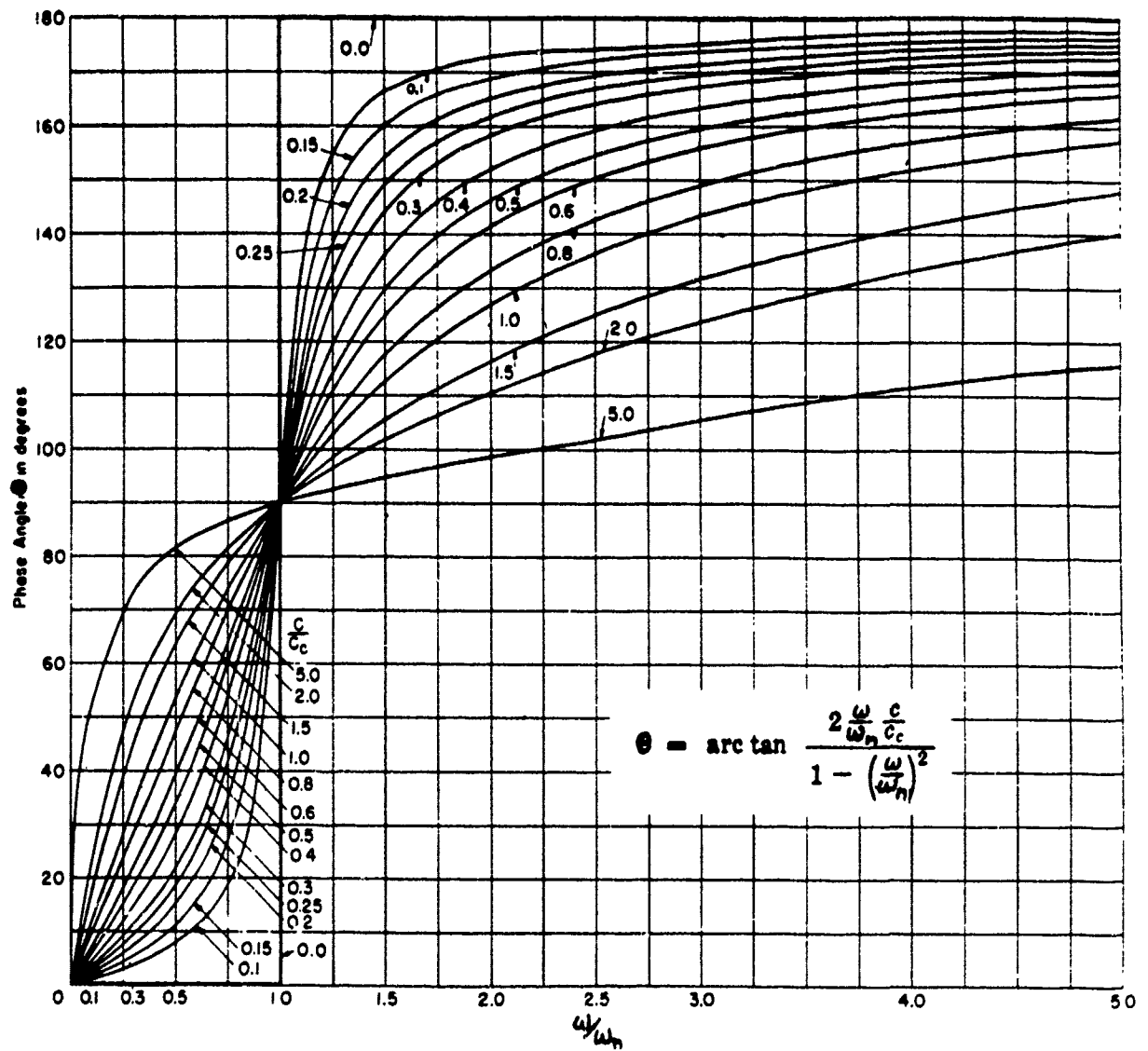


Figure A-11. Phase Angle Plotted Against ω/ω_n with c/c_c as a Parameter for a System of One Degree-of-Freedom Acted on by a Force $P_0 \sin \omega t$

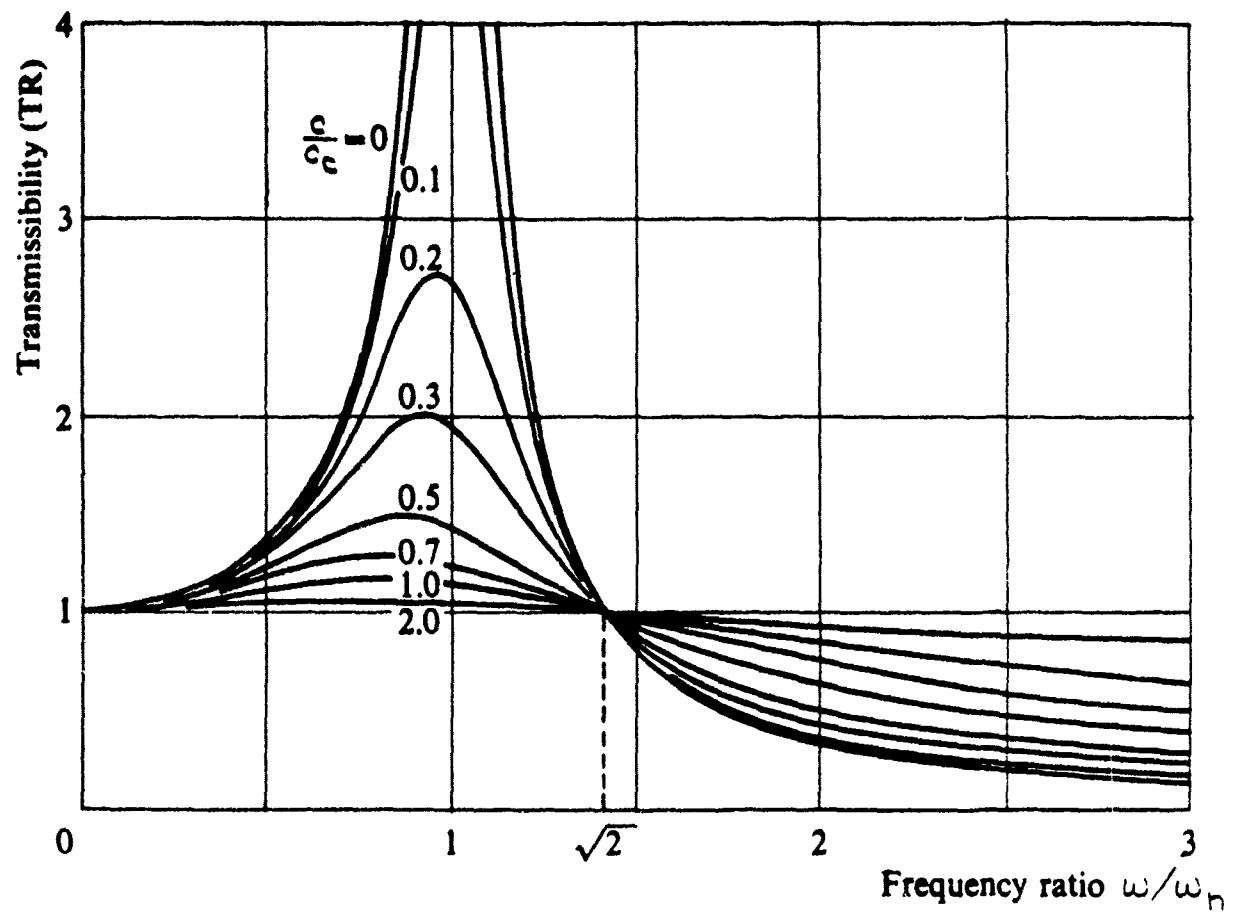


Figure A-12. Transmissibility of a Viscous Damped, Forced, Single Degree-of-Freedom System Plotted Against ω/ω_n for Various Values of c/c_c

It is defined as

$$Q = \frac{1}{2\frac{c}{c_c}}$$

and from the previous expression for x_o , it follows that when $\omega = \omega_n = \sqrt{\frac{k}{m}}$,

$$F = \frac{x_o}{x_{st}} = \frac{1}{2\frac{c}{c_c}}$$

Q can be approximated from a plot of the magnification factor if the damping is small ($c/c_c \leq 0.1$). See Figure A-13. If Δf is the frequency increment where x_o is at or above the half-power point

$$\left(x_o > \frac{\text{Max } x_o}{\sqrt{2}} \right)$$

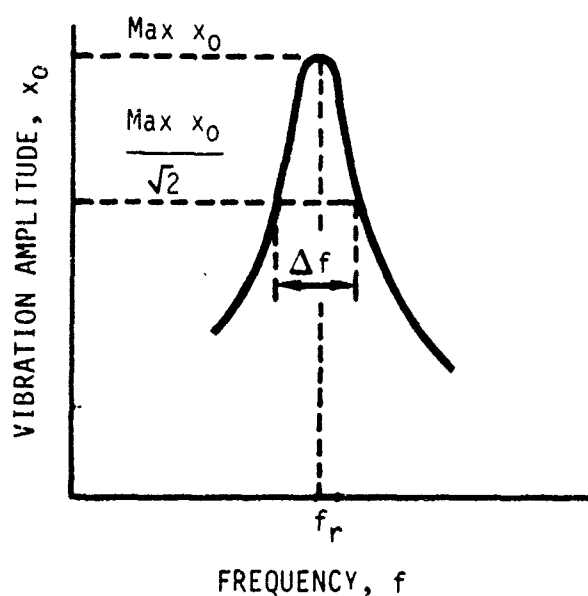


Figure A-13. Approximating Q from Response Curve

and f_r is the resonant frequency, then

$$Q \approx \frac{f_r}{\Delta f}$$

If damping is due to a constant friction force (coulomb damping) it may be convenient to use, as an approximation, an equivalent viscous damping constant, c , which would dissipate an equal amount of energy.

$$c = \frac{4F_c}{\pi\omega x_0}$$

where x_0 = amplitude of vibration, in.

ω = circular frequency, $\frac{\text{radians}}{\text{sec}}$

F_c = friction force, lbs.

c = equivalent linear viscous damping constant, $\frac{\text{lb-sec}}{\text{in}}$

A.2.3 Multiple Degree-of-Freedom Systems

A mechanical system may have to be represented by an arrangement with more than one mass, or with a mass moving in more than one direction. A beam can be considered as an arrangement of an infinite number of masses; this is called continuous distribution.

The number of independent coordinates required to describe the position of the system is the number of degrees-of-freedom of that system. For example, if a system contains four masses, each one of which is free to move only in the x and y directions, the system has eight degrees-of-freedom. It is possible for a single rigid body to have six degrees-of-freedom; three translational and three rotational.

To describe the motion of a multiple degree-of-freedom system an equation of forces acting on each mass in each direction can be written. In a system with N degrees-of-freedom, there will be N differential equations with N variables to describe the motion.

For example, the motion of the mechanical system of three degrees-of-freedom shown in Figure A-14 can be described by:

$$\begin{aligned} m_1 \ddot{x}_1 + k_1 x_1 - k_2 (x_2 - x_1) &= P_0 \sin \omega t \\ m_2 \ddot{x}_2 + k_2 (x_2 - x_1) - k_3 (x_3 - x_2) - c \dot{x}_2 &= 0 \\ m_3 \ddot{x}_3 + k_3 (x_3 - x_2) - k_4 x_3 &= 0 \end{aligned}$$

A system free from external force will oscillate only at certain frequencies which are the natural frequencies of the system. There are as many natural frequencies as there are degrees-of-freedom in a system. It is possible for a system to vibrate at any one or combination of its natural frequencies.

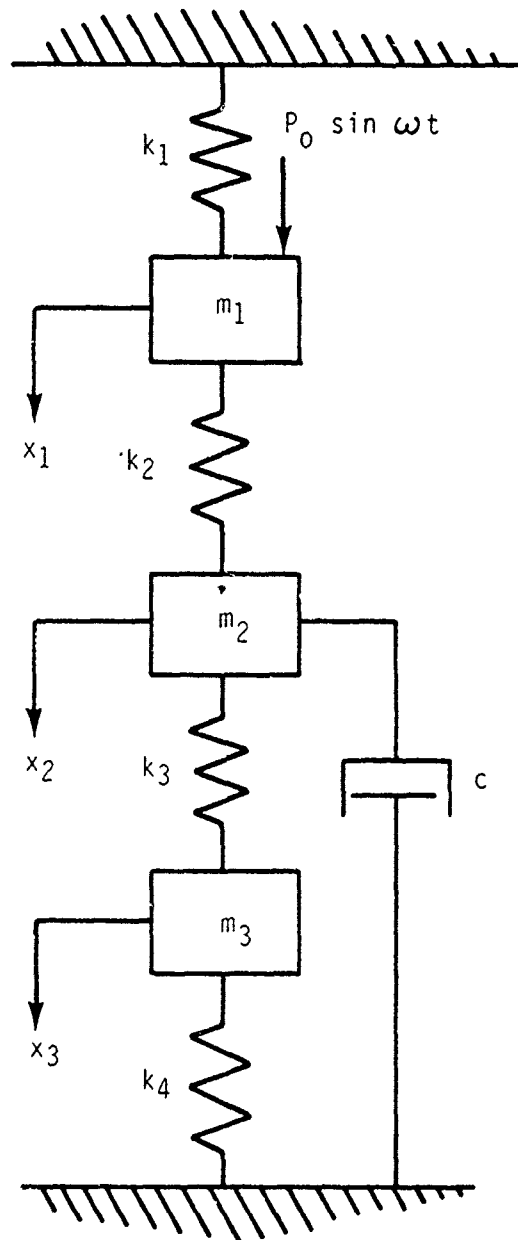


Figure A-14. Example of a Three Degree-of-Freedom System

A structure vibrating at only one of its natural frequencies (in one of its modes) does so with a characteristic pattern of amplitude distribution called a mode shape.

A.3 VIBRATION CONTROL

The vibration present in mechanical systems may be forced (non-resonant) or resonant. Typical cases of forced vibration aboard ship include hull vibration excited by propeller blade frequency, the number of propeller blades times the shaft rpm, when the exciting frequency is above the sixth mode natural frequency of the hull girder. In this case, resonance does not occur and the vibration of the hull girder is directly related to the transmitted forces. If the resulting vibration is excessive, the solution of the problem requires the reduction of the exciting forces.

Similarly, vibration of a machine may be nonresonant and the response may increase in service due to unbalance or worn bearings. Machine vibration is commonly monitored to determine requirements for maintenance, at which time the forces generated are reduced to satisfactory levels. In such cases, the solution to problems of forced vibration is limited to the reduction of the exciting forces.

Resonant vibration occurs when the frequency of the excitation is equal to a natural frequency of the system. When this happens, the amplitude of vibration will increase and is only limited by the amount of damping present in the system. Therefore, in order to avoid disastrous effects resulting from very large amplitude of vibration at resonance, the natural frequency of a system must be known and properly taken care of.

In theory, the treatment of resonance vibration provides greater flexibility for solution of a vibration problem in that the exciting force may be reduced; the frequency of the exciting force may be altered, such as by changing the number of propeller blades in the case of hull vibration; the natural frequency of the mechanical system may be changed by changing the mass and/or stiffness constants; or by the introduction of system damping. Typical cases frequently encountered are the longitudinal vibration of the main propulsion system of geared turbine drives and the torsional vibration of diesel driven ships. It is considered mandatory that such systems be completely analyzed and evaluated for satisfactory performance during the design stage.

When investigating shipboard vibration it is most important to determine if the observed vibration is forced or resonant. This can be done by altering the frequency of the excitation, by changing the RPM of the propeller or machine associated with the problem. If the system is at, or near, resonance, a significant change in amplitude of vibration will be noted with the change in frequency. As noted above, if it is concluded that the vibration is non-resonant and results from excessive forces, your options are limited. While your options are greater in the treatment of resonant conditions, practical considerations may still restrict, or limit, your approach to a ready solution. To avoid such situations it is necessary to analyze the mechanical system, preferably during the design stage, to minimize the occurrence of serious problems.

A.4

NATURAL FREQUENCIES OF COMMON SYSTEMS

Often shipboard equipment can be represented by a simplified mechanical system for purposes of calculating its natural frequency(ies). In solving vibration problems these simplified models can be used to determine the effect of changing the stiffness or mass of the equipment or its support. When raising or lowering the natural frequency to avoid resonances, the existing system natural frequency should be measured and calculated first to verify the validity of the model. Table A-4 gives formulas for natural frequencies. Tables are also included for the stiffness of some common structures (Table A-5) and moments of inertia of some cross sections (Table A-6). Notations for all three tables is given in Table A-7.

Table A-4. Natural Frequencies of Common Systems

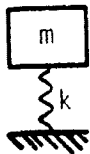
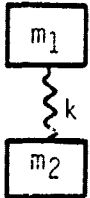
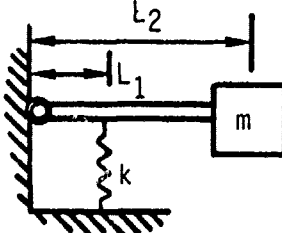
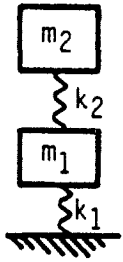
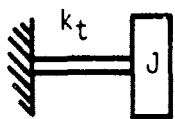
| MODEL | SKETCH | NATURAL FREQUENCY, HZ |
|-----------------------------|---|--|
| 1. Single mass and spring |  | $f_n = \frac{1}{2\pi} \sqrt{\frac{k}{m}}$ |
| 2. Two masses, one spring |  | $f_n = \frac{1}{2\pi} \sqrt{\frac{k(m_1 + m_2)}{m_1 m_2}}$ |
| 3. Single cantilevered mass |  | $f_n = \frac{L_1}{2\pi L_2} \sqrt{\frac{k}{m}}$ |

Table A-4. Natural Frequencies of Common Systems (Continued)

| MODEL | SKETCH | NATURAL FREQUENCY, HZ |
|----------------------------|---|--|
| 4. Two mass, two spring |  | $f_n = \frac{1}{4\pi} \left[\frac{k_1}{m_1} + \frac{k_2}{m_2} \left(1 + \frac{m_2}{m_1} \right) \pm \sqrt{\left[\frac{k_1}{m_1} + \frac{k_2}{m_2} \left(1 + \frac{m_2}{m_1} \right) \right]^2 - \frac{4k_1 k_2}{m_1 m_2}} \right]$ |

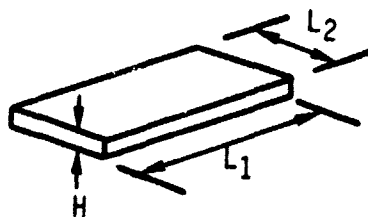
5. One rotor and
shaft



$$f_n = \frac{1}{2\pi} \sqrt{\frac{k_t}{J}}$$

NOTE: Similar to Model No. 1. Two rotor systems are similar to Model No. 2 or 4.

6. Rectangular plate,
simply supported
edges
(no constraints)

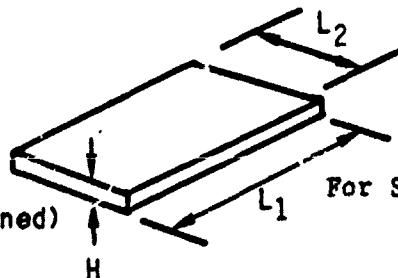


$$f_n = 8.91 H \left(\frac{1}{L_1^2} + \frac{1}{L_2^2} \right) \sqrt{\frac{E}{(1 - \nu^2)\rho}}$$

For Steel:

$$f_n = 9.7 \times 10^4 H \left(\frac{1}{L_1^2} + \frac{1}{L_2^2} \right)$$

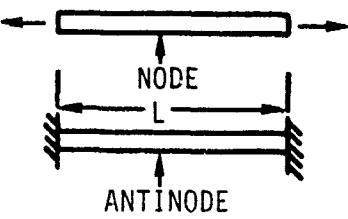
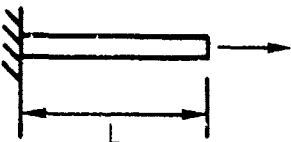
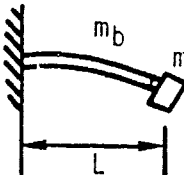
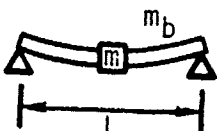
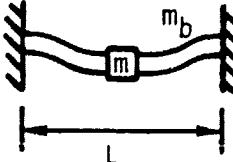
7. Rectangular plate,
clamped edges
(all edges constrained)



For Steel:

$$f_n = 8.2 \times 10^4 \frac{H}{L_1 L_2} \sqrt{7 \left(\frac{L_1^2}{L_2^2} + \frac{L_2^2}{L_1^2} \right) + 4}$$

Table A-4. Natural Frequencies of Common Systems (Continued)

| MODEL | SKETCH | NATURAL FREQUENCY, HZ |
|--|---|---|
| 8. Longitudinal vibration of beam (fixed or free ends) |  | $f_n = \frac{9.82}{L} \sqrt{\frac{E}{\rho}}$ <p>For Steel or Aluminum:</p> $f_n = \frac{10^5}{L}$ |
| 9. Longitudinal vibration of beam (fixed-free) |  | $f_n = \frac{1.57}{2\pi} \sqrt{\frac{AE}{\mu L^2}}$ |
| 10. Fixed-Free beam mass at end |  | $f_n = \frac{1}{2\pi} \sqrt{\frac{3EI}{L^3 (m + .23 m_b)}}$ |
| 11. Hinged-Hinged beam mass in center |  | $f_n = \frac{1}{2\pi} \sqrt{\frac{48EI}{L^3 (m + .5 m_b)}}$ |
| 12. Fixed-Fixed beam mass in center |  | $f_n = \frac{7}{\pi} \sqrt{\frac{EI}{L^3 (m + .375 m_b)}}$ |

NOTE: For Models 10-12, set $m_b = 0$ for massless beams.

Table A-4. Natural Frequencies of Common Systems (Continued)

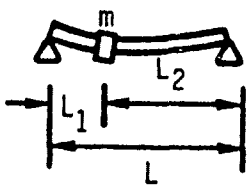
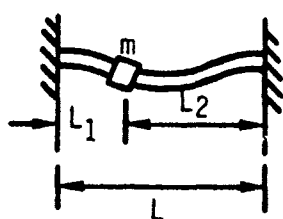
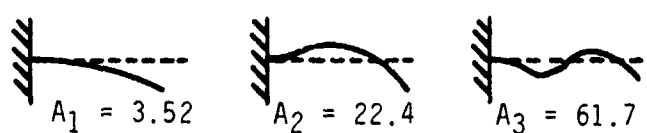
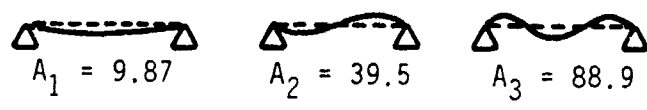
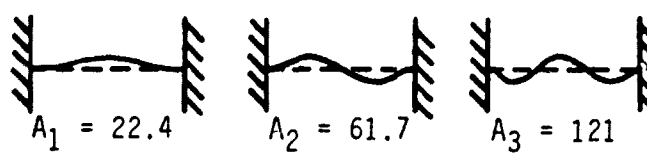
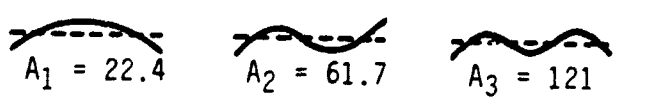
| MODEL | SKETCH | NATURAL FREQUENCY, HZ |
|--|--|---|
| 13. Hinged-Hinged beam, off-center mass, massless beam |  | $f_n = \frac{1}{2\pi L_1 L_2} \sqrt{\frac{3EI L}{m}}$ |
| 14. Fixed-Fixed beam, off-center mass, massless beam |  | $f_n = \frac{1}{2\pi} \sqrt{\frac{3EI L^3}{m L_1 L_2}}$ |
| 15. Uniform beam, fixed-free |  | $f_1 = \frac{A_1}{2\pi} \sqrt{\frac{EI}{\mu L^4}}$ |
| 16. Uniform beam, hinged-hinged |  | $f_1 = \frac{A_1}{2\pi} \sqrt{\frac{EI}{\mu L^4}}$ |
| 17. Uniform beam, fixed-fixed |  | $f_1 = \frac{A_1}{2\pi} \sqrt{\frac{EI}{\mu L^4}}$ |
| 18. Uniform beam, free-free |  | $f_1 = \frac{A_1}{2\pi} \sqrt{\frac{EI}{\mu L^4}}$ |

Table A-4. Natural Frequencies of Common Systems (Concluded)

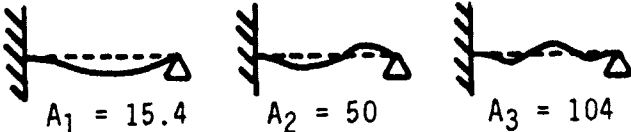
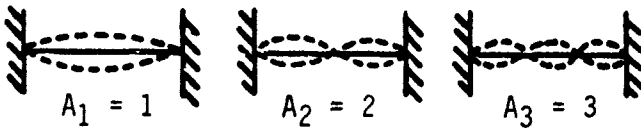
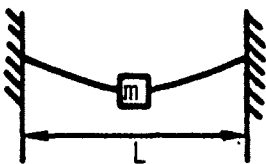
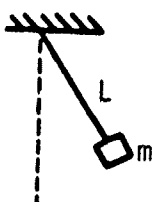
| MODEL | SKETCH | NATURAL FREQUENCY, HZ |
|-------------------------------------|---|--|
| 19. Uniform beam, fixed-hinged |  | $f_1 = \frac{A_1}{2\pi} \sqrt{\frac{EI}{\mu L^4}}$ |
| 20. Uniform string in tension |  | $f_1 = \frac{A_1}{2L} \sqrt{\frac{P_s}{\mu}}$ |
| 21. Massless string, mass in center |  | $f_n = \frac{1}{\pi} \sqrt{\frac{P_s}{mL}}$ |
| 22. Pendulum |  | $f_n = \frac{1}{2\pi} \sqrt{\frac{g}{L}}$ |

Table A-5. Stiffnesses of Common Structures

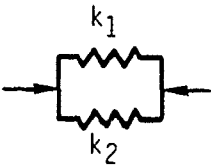

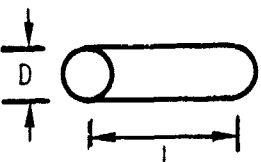
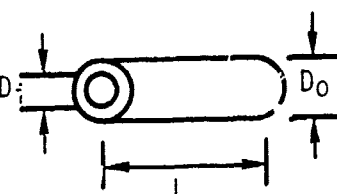
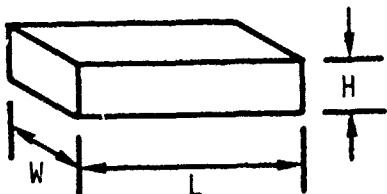
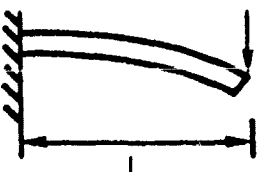
| MODEL | SKETCH | STIFFNESS |
|---------------------------|---|--|
| 1. Parallel springs |  | $k = k_1 + k_2$ |
| 2. Series springs |  | $k = \frac{k_1 k_2}{k_1 + k_2}$ |
| 3. Solid shaft (torsion) |  | $k_t = \frac{\pi G D^4}{32 L}$ |
| 4. Hollow shaft (torsion) |  | $k_t = \frac{\pi G (D_o^4 - D_i^4)}{32 L}$ |
| 5. Rectangular (torsion) |  | $k_t = \frac{G W H^3}{3 L}$ |
| 6. Cantilever beam |  | $k = \frac{3 E I}{L^3}$ |

Table A-5. Stiffnesses of Common Structures (Concluded)

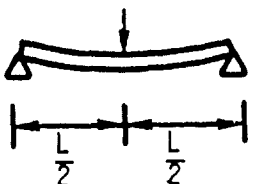
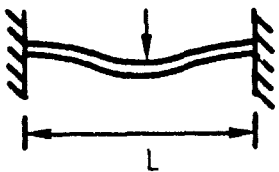
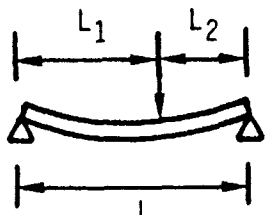
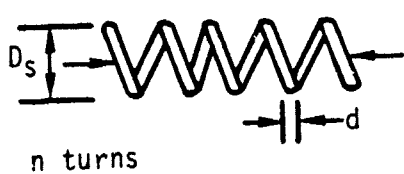
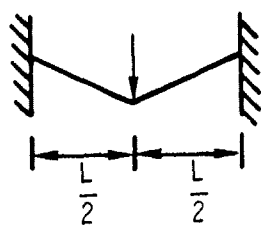
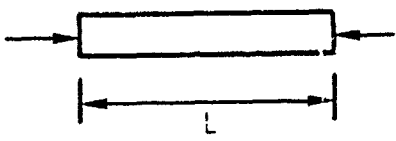
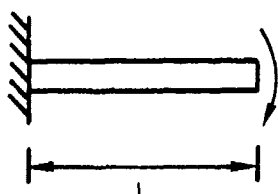
| MODEL | SKETCH | STIFFNESS |
|--|--|--------------------------------|
| 7. Simply supported beam (center load) |  | $k = \frac{48EI}{L^3}$ |
| 8. Fixed-Fixed beam |  | $k = \frac{192EI}{L^3}$ |
| 9. Simply supported beam (off-center load) |  | $k = \frac{3EIL}{L_1^2 L_2^2}$ |
| 10. Helical spring |  | $k = \frac{Gd^4}{8nD_s^3}$ |
| 11. String in tension |  | $k = \frac{4Ps}{L}$ |
| 12. Axial stiffness of beam |  | $k = \frac{AE}{L}$ |
| 13. Torsional stiffness of beam |  | $k_t = \frac{GI}{L}P$ |

Table A-6. Moments of Inertia of Some Common Cross Sections

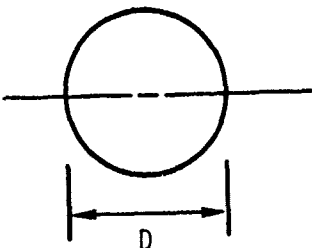
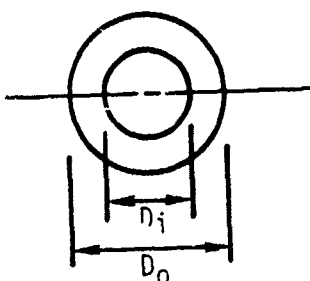
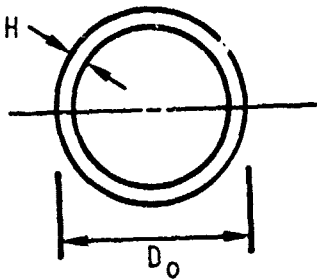
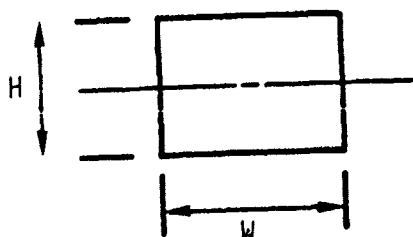
| <u>CROSS-SECTION</u> | <u>SKETCH</u> | <u>I (WITH RESPECT TO AXIS SHOWN) OR I_p</u> |
|----------------------|--|---|
| Circle |  | $I = \frac{\pi D^4}{64}$ $I_p = \frac{\pi D^4}{32}$ |
| Hollow Circle |  | $I = \frac{\pi}{64} (D_o^4 - D_i^4)$ $I_p = \frac{\pi}{32} (D_o^4 - D_i^4)$ |
| Thin Wall Circle |  | $I = \frac{\pi D_o^3 H}{8}$ $I_p = \frac{\pi D_o^3 H}{4}$ |
| Rectangle |  | $I = \frac{WH^3}{12}$ |

Table A-7. Notation for Tables 4-4, 4-5, and 4-6

| | |
|--------|--|
| m | Mass, lb-sec ² /in, weight/g |
| k | Translational stiffness, lb/in |
| k_t | Torsional stiffness, lb-in/rad |
| J | Mass moment of inertia, lb-in-sec ² |
| E | Young's Modulus, lbs/in ² |
| ν | Poisson's Ratio $\left(G = \frac{E}{2(1 + \nu)}\right)$ |
| ρ | Weight density, lbs/in ³ |
| m_b | Mass of beam, lb-sec ² /in |
| I | Area moment of inertia of cross-section, in ⁴ |
| I_p | Area polar moment of inertia of cross-section, in ⁴ |
| L | Length of beam or string, in |
| μ | Mass per unit length of beam or string, lb-sec ² /in ² |
| P_s | Tension in string, lb |
| A | Cross-section area of beam, in ² |
| g | Acceleration due to gravity, 386.1 in/sec ² |
| G | Shear modulus of elasticity, lb/in ² |
| I_p | Polar area moment of inertia of cross-section |

A.5 REFERENCES

- A-1. ASA S1.1 - 1960, "Acoustical Terminology (Including Mechanical Shock and Vibration)."
- A-2. Den Hartog, J.P., "Mechanical Vibrations," 2nd ed., McGraw-Hill Book Company, Inc., New York, 1940.
- A-3. Harris, C.M., and C.E. Crede, "Shock and Vibration Handbook," McGraw-Hill Book Company, Inc., New York, 1961.
- A-4. McGoldrick, R.T., "A Vibration Manual for Engineers," 2nd ed., TMB Report R-189, December 1957.
- A-5. McGoldrick, R.T., "Ship Vibration," TMB Report 1451, December 1960.

Appendix B

REFERENCE DOCUMENTS

In the preparation of this guide reference is made, whenever possible, to Codes, Standards, Specifications, and Design Guides applicable to the evaluation and control of shipboard vibration. The principal sources include publications of professional, classification, and standards groups, as well as publications of the U.S. Navy and its laboratories. Specific references throughout the text are given in each section. The most important reference documents are listed here in two categories:

- Related Standards and Specifications
- Technical References

The first category of related standards and specifications provide detailed information on the measurement and evaluation of shipboard vibration, while the second category includes some selected publications which deal with ship design and construction, and includes much background information which has contributed to the development of the present standards and specifications.

B.1 RELATED STANDARDS AND SPECIFICATIONS

B.1.1 International Organization for Standardization - Pertinent Documents

| | |
|--------------|---|
| ISO 2041 | Vibration and Shock Vocabulary, 1975 |
| ISO/DIS 4865 | Vibration and Shock - Methods for the Analysis and Presentation of Data |
| ISO 4867 | Code for the Measurement and Reporting of Shipboard Vibration Data, 1983 |
| ISO 4868 | Code for the Measurement and Reporting of Shipboard Local Vibration Data, 1983 |
| ISO/DIS 6954 | Mechanical Vibration and Shock - Guidelines for the Overall Evaluation of Vibration in Merchant Ships, 1983 |
| ISO 2631 | Guide for the Evaluation of Human Exposure to Whole Body Vibration, 1978 |

B.1.2 Society of Naval Architects and Marine Engineers (SNAME)

| | |
|-------|--|
| SNAME | T&R Code C-1, Code for Shipboard Vibration Measurement, 1975 |
| SNAME | T&R Code C-4, Shipboard Local Structure and Machinery Vibration Measurements, 1976 |

SNAME T&R Code C-5, Acceptable Vibration of Marine Steam
and Heavy Duty Gas Turbine Main and Auxiliary
Machinery Plants, 1976

B.1.3 U.S. Navy Department, Naval Sea Systems Command

MIL-STD-167 (SHIPS) Mechanical Vibrations of Shipboard Equipment,
1974

Type I - Environmental
Type II - Internally Excited

(Reciprocating Machinery, Propulsion System,
and Shafting)

Type III - Torsional Vibration
Type IV - Longitudinal Vibration
Type V - Lateral (Whipping) Vibration

MIL-STD-1472B Human Engineering Design Criteria for
Military Systems, Equipment and Facilities,
1974

B.2 TECHNICAL REFERENCES

B.2.1 Society of Naval Architects and Marine Engineers (SNAME)

Books

Marine Engineering, 1971, Edited by Harrington
Ship Design and Construction, 1980, Edited by Taggart
Principles of Naval Architecture, 1967, Edited by Comstock
Hydrodynamics in Ship Design, Vols. I & II, 1957, Edited by
Saunders
Hydrodynamics in Ship Design, Vol. III, 1965, Edited by Saunders

Technical Bulletins

T&R Bulletin 2-25, Ship Vibration and Noise Guidelines, 1980
Techniques for Estimating Vibratory Forces Generated by Propellers,
1975, 54 pages, 1-34
Guide to Propulsion Reduction Gear Alignment and Installation,
1961, 76 pages, 3-10

Data Sheets

Vibration Data Sheets, Set of 5, 1976, D-14
Vibration Data Sheets, Set of 5, 1977, D-15

Technical and Research Reports

Longitudinal Stiffness of Main Thrust Bearing Foundations, 1972,
22 pages, R-15
Results of a Survey on Shaft Alignment Procedures Used by American
Shipyards, 1978, 12 pages, R-25

National Symposia

Ship Structure Symposium, 1975, SY-5
Ship Vibration Symposium, 1978, SY-8

B.2.2

Classification Societies

Rules for Building and Classing Steel Vessels, American Bureau of
Shipping (ABS)
Prevention of Harmful Vibration in Ships, Det Norske Veritas
(DNV), May 1980
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